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President, D. O. Price
Vice-President, W. C. Kelly
Secretary-Treasurer, H. R. Roth
Board of Governors, E. R. Gauley, H. G. Hill, A. S. Morgan, C. Tasker

Oregon

Headquarters, Portland, Ore.
Meets: Thursday after First Tuesday

President, B. W. Farnes
Vice-President, J. A. Freeman
Secretary, R. E. Chase, Jr., G. H. Risley *
Treasurer, F. F. Urban
Board of Governors, W. R. Burton, J. F. McIndoe, W. J. Widmer

OFFICERS OF LOCAL CHAPTERS—1942 (*continued*)

Pacific Northwest

Headquarters, Seattle, Wash.
Meets: Second Tuesday

President, H. T. Griffith
Vice-President, R. D. Morse
Secretary, R. E. Le Riche
Treasurer, G. E. Mead, E. H. Langdon *
Board of Governors, M. J. Hauan, F. J. Pratt, R. O. Wesley

Philadelphia

Headquarters, Philadelphia, Pa.
Meets: Second Thursday

President, H. H. Mather
First Vice-President, Edwin Elliot
Second Vice-President, A. C. Caldwell
Secretary, M. G. Kershaw
Treasurer, R. D. Touton
Board of Governors, E. H. Dafter, H. B. Hedges

Pittsburgh

Headquarters, Pittsburgh, Pa.
Meets: Second Monday

President, C. M. Humphreys
Vice-President, G. G. Waters
Secretary, E. H. Riesmeyer, Jr.
Treasurer, L. S. Maehling
Board of Governors, T. F. Rockwell, E. C. Smyers, H. B. Steggall

St. Louis

Headquarters: St. Louis, Mo.
Meets: First Tuesday

President, M. F. Carlock
First Vice-President, C. F. Boester
Second Vice-President, J. H. Carter
Secretary, W. J. Oonk
Treasurer, B. C. Simons
Board of Governors, B. L. Evans, D. J. Fagin, G. B. Rodenheiser, E. H. Ronsick

South Texas

Headquarters, Houston, Tex.
Meets: Third Friday

President, D. S. Cooper
Vice-President, A. J. Mitchell
Secretary-Treasurer, A. M. Chase, Jr.
Board of Governors, A. B. Banowsky, W. R. Etie, A. J. Rummel

* Filled unexpired term.

Southern California

Headquarters, Los Angeles, Calif.
Meets: Second Wednesday

President, H. H. Bullock
Vice-President, W. O. Stewart
Secretary, Leo Hungerford
Treasurer, Maron Kennedy
Board of Governors, J. B. Griffith, A. J. Hess, P. C. Scofield, E. B. Wells

Washington, D. C.

Headquarters, Washington, D. C.
Meets: Second Wednesday

President, R. S. Dill
Vice-President, F. M. Thuney
Secretary, J. W. Markert
Treasurer, F. B. Sale
Board of Governors, S. L. Gregg, F. A. Leser, W. H. Littleford

Western Michigan

Headquarters, Grand Rapids, Mich.
Meets: Second Monday

President, F. C. Warren
Vice-President, C. H. Pesterfield
Secretary, H. D. Bratt
Treasurer, V. H. Hill
Board of Governors, W. C. DeRoo, H. J. Metzger, W. G. Schlichting

Western New York

Headquarters, Buffalo, N. Y.
Meets: Second Monday

President, H. C. Schafer
First Vice-President, S. M. Quackenbush
Second Vice-President, S. W. Strouse
Secretary, Herman Seelbach, Jr.
Treasurer, B. C. Candee
Board of Governors, M. C. Beman, Joseph Davis, Roswell Farnham, W. R. Heath, D. J. Mahoney

Wisconsin

Headquarters, Milwaukee, Wis.
Meets: Third Monday

President, H. W. Schreiber
Vice-President, F. W. Goldsmith
Secretary, C. S. Becker, I. J. Haus *
Treasurer, O. A. Trostel
Board of Governors, T. M. Hughey, C. W. Miller, C. H. Randolph

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TRANSACTIONS

of

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 1191

FORTY-EIGHTH ANNUAL MEETING, 1942
Philadelphia, Pa.

THE 48th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS opened at the Bellevue-Stratford Hotel, Philadelphia, January 26 with a business session for members and Pres. W. L. Fleisher called the meeting to order at 10:00 A.M. He introduced M. F. Blankin, General Chairman of the Committee on Arrangements, for the Philadelphia Chapter who presented the Hon. Charles H. Grakelow, Director of Purchases and Supplies of the City of Philadelphia, who on behalf of Mayor Bernard Samuel, presented greetings and a warm welcome to the city. He stressed the importance of the city's contribution to the National Defense Program and its wealth of valued historical inheritances from the early days of the founding of the country. He expressed the hope that members would take advantage of the opportunity, "to visit the Cradle of Liberty and stand by that hallowed Bell whose metal tongue has long since been silent and yet which wields an influence the like of which I cannot describe, as evidenced by the men and women who come from all sections of America to kneel in adoration and homage and gratitude for all it has meant to us in these years."

President Fleisher responded briefly on behalf of the Society Members. The personnel of the Resolutions Committee was then announced by President Fleisher as follows: Lester T. Avery, Cleveland, *Chairman;* C. E. Bentley, San Francisco, and Brig.-Gen. W. A. Danielson, Memphis.

President Fleisher introduced Arthur W. Nass, President of the *Heating, Piping and Air Conditioning Contractors National Association,* who brought greetings from his members and presented an interesting discussion about the relationship between the engineer and the contractor. He briefly outlined some of the problems involved and expressed the thought that a frank discussion of these matters was desirable.

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President's Address

It is customary, when a President has completed his term in office, to review the important things that have transpired during his incumbency, to record what was accomplished and what can be done for the future development of the Society.

There is no need to tell you that 1941 was a trying year. The actual entry of the United States into the war, in December, had been so long anticipated, that only the tragedy of Pearl Harbor made it anything but an anti-climax. As a result of my visits to Europe in 1938 and 1939, it was my conviction that it would be impossible for the United States to stay out of a world war. Consequently, on my frequent appearances before the chapters throughout the country and before various other organizations and agencies, I stressed the necessity for adequate defense preparations and particularly civilian air raid protection, predicated primarily on ventilation of shelters.

While formerly the President of the Society made visits to the chapters in a continuous tour, during his term, lasting for about three weeks, the increase in the number of our chapters and the other arduous responsibilities of the President made it essential to spread these visits over a longer period of time, so in 1939 the First and Second Vice-Presidents were assigned certain of these duties, in order to relieve the President. This plan has proved a very satisfactory method of distributing the duties of the executive officers of the Society and, I believe, should be continued. My own tours started some three years ago, and, according to my diary, ever since then I have been traveling extensively in the interests of the Society.

It has been my endeavor to focus attention upon the value of research and the educational importance of the Society, not only throughout the United States and Canada, but also in England, France, Belgium and Holland.

Upon entering the Presidency of the Society, I made a detailed study of the Society's fundamental policies and how its influence could be furthered and its technical investigations and research developed for the education of the public and the enhancement of its position in national affairs. This is something that must be very carefully considered, because particularly during the past year, when the government has been drawing from the various technical societies for advisory boards, the Society, in my opinion, has been overlooked and neglected, because it has not been considered one of the founder societies. The actual importance of the Society is so great, that it is only a lack of the understanding of its accomplishments that has put it in this position.

In order to carry through those things which might be considered essential for the advancement of the Society, I prepared and presented to the Council in April a statement of policy, which, unfortunately, (Russia not having come into the war as an ally) I called a five-year plan. A better name might have been chosen, but this plan which is on record in the Minutes of the Council, divided the work of the Society into those branches which I considered most important for its development and life:

1st— Research
2nd— Publications (I refer to the monthly publications), our GUIDE and something to take the place of the TRANSACTIONS
3rd— Education—that is, education of the members, education of the professions and education of the layman
4th— Codes and Standards to set the rules and guide the courts and municipalities
5th— Publicity to advertise the work of the Society in research, education and standards.

In this report research was emphasized—research as developed by the Society over its 20 odd years of investigation and the development of new research projects, which, to some members might seem inappropriate and out of place in the heating, ventilating and air conditioning field. The three matters that I stressed most particularly in this report were:

1st— the spread of our activities to include those things that actually are in line with our basic knowledge, such as those parts of aeronautics that might be considered air movement or air control, having to do primarily with the testing and not the design of planes.
2nd— continued stress on the physiological and medical aspects of air or air conditioning.
3rd— the matter of the method of publication of our research and educational work.

Another important part of the plan for the future welfare of the Society was the very thing which at this particular moment is threatening to jeopardize the research of the Society, namely, *money*, for carrying on research. This should be taken out of the realm of instability and I suggest that every means be adopted to create an endowment which will carry our research forward over trying periods. If necessary, I advocate that money be taken from the surplus of the Society to continue our research on an adequate scale during the war period and the depression which is sure to follow the war.

I wish to take this opportunity of expressing my appreciation of the tremendous assistance that your Secretary, A. V. Hutchinson, has contributed in carrying on the affairs of the Society. But I want to disabuse the minds of the members—and this I know I do with the full concurrence of Mr. Hutchinson—of the idea that the Secretary can run the Society. The tremendous duties that are imposed upon him by the publication of THE GUIDE, the general business of the Society and the innumerable calls on his time and energy leave him no time to develop or establish policies, nor to decide or pass upon the scientific activities with which the Society is most nearly concerned. The commendable work of the Technical Secretary, John James, and of Director F. C. Houghten of the Laboratory, should also be appreciated by the members.

In a society such as this, the executive officers should be scientifically minded and interested in research, and education, and the head of the Society not only must be familiar with all of these activities, but must be competent to pass upon any critical points that may arise in connection with them. Moreover, he is expected to understand showmanship almost as well as Barnum and to assist in the success of the expositions which may be held while he is in office. This year I had the unique experience of participating in the preparations for two expositions, one in San Francisco in June and the currently postponed exposition in Philadelphia at this time. Both of these took considerable time, which ordinarily the President of the Society is not called upon to give. It is also the duty of the President, if he is interested in research, to see how money can be obtained to support this work, and the contacts that he can make are very important. While a fine record has been made in this work by E. P. Heckel of the Finance Committee of Research, it has been necessary for me to devote considerable time in the last few months in an attempt to create interest in new research and also secure the necessary funds to conduct the work.

The past year has been a difficult one for your President for many reasons, as both the last Annual Meeting in Kansas City, and the Summer Meeting in San Francisco required unusually long absences from home and office. Also the necessity of conserving the funds of the Society made it essential for the President to avail himself of his locality to bring news of the Society to adjacent chapters.

These items are cited so that you may realize that although a man is greatly honored in being elected to this high office, his time and his duties sometimes impose burdens that the executive officers of the Society must be willing to contribute and be able to afford. One who serves as President will find that there is no week and practically no day in the entire year when some time must not be devoted to the work of the Society. This was my experience during the past year.

As one of my foremost interests has been in Society research for the last ten years, it was my good fortune to have Commander A. E. Stacey, Jr. accept reappointment as Chairman of the Committee on Research, and he devoted a large share of his time to this important work until he was called on by his government to enter active service in the U. S. Navy, during the latter part of 1941. Dr. Winslow, Vice-Chairman, was called to the west coast for a six-months' assignment, so he was handicapped somewhat in assisting the chairman. J. H. Walker, the Committee's Technical Adviser, had been on the sick list for many months but, fortunately, he could be active in the last few months and he has done an excellent job.

Consequently, your President has felt it his duty to keep in closer touch with research affairs than is usually customary for the President.

None of the problems of the Presidency have been mentioned with the idea of arousing your sympathy or soliciting your commendation. If I had not loved everything pertaining to the Society, if I were not devoted to research, if I did not feel that we had an opportunity to advance the comfort and health of humanity as no

other organization has ever done, I undoubtedly would have found a way out of many of the arduous duties which the office imposes necessarily. But when one realizes the part that we as a Society and as individual members of this Society have been able to accomplish for the good of the country in the national emergency with which we are now entangled, one appreciates the joy with which one looks back on his association with this group.

During the October Council meeting in Washington, I tried to summarize the contribution of the Society to national defense, and it was a revelation to find how in practically everything entering into the conduct of the war the research of the Society has contributed considerably to the development of the implements of war and the projection of the people.

Our work over the past 20 years is crystalizing now in indicating what conditions of temperature and humidity are required for the maximum output of office workers, engineers and draftsmen, and the proper manufacture of war material. Blackout buildings have made the requirements for ventilation so important, that the work we have done on air conditioning requirements in industry is the basis of all ventilation design.

Our work in fever therapy is having its effect on the medical treatment of social diseases. Our work in operating rooms and convalescent wards is the basis of the proper design for carrying on the emergency work in hospitals, unfortunately so essential in the coming months.

Our research in spot cooling and proper clothing and the design of ventilation for hot spaces in ships and cold surroundings in stratosphere flying is being utilized by the Army and Navy.

Airplane plants, airplane body plants and an infinite number of other industrial institutions are profiting from our research work.

Our new cantonments are utilizing our research in heating, ventilating and air filtering.

There is practically not a single phase of production for defense in which the work of the Society is not playing a prominent part.

I am more than happy to have had the opportunity, at the conclusion of my long period of close association with the work of the Society, to have served as your President and executive head during this time of national emergency.

Respectfully submitted,
W. L. FLEISHER, *President*

The Secretary was requested to present a report of the Council which he read as follows:

Council Report

The Council held five meetings during the past year and carried out its duties under the Constitution and By-Laws and acted on all special matters which required its attention.

The Council held its organization meeting January 29, 1941, in Kansas City, Mo., with Pres. W. L. Fleisher presiding and there were 16 of the 18 members present. Announcement of Committee appointments was made and the Society's Budget for 1941, providing for $85,600 expense exclusive of Research was adopted. A Certified Public Accountant was appointed and depositories for Society funds were selected. The Secretary and Technical Secretary were reappointed.

The April Council Meeting was held in Toronto and a grant of $5000 from Society funds was made to the Committee on Research. The reports of Special Committees were received and the Program for the Summer Meeting in San Francisco was approved. President Fleisher submitted a program for the Promotion and Development of the Society which was discussed in great detail and these sections relating to publications were referred to a Special Committee; the topics covering educational work were referred to the Local Chapters Committee and the material bearing on Research was sent to the Committee on Research for study.

In June, the Council met in San Francisco and authorized the appointment of a Committee to cooperate with the *ASRE* and a Special Committee to study the proposal for publication of weather data. President Fleisher announced the appointment

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of a Special Committee on Medical and Engineering Research; an appropriation was made for the employment of special personnel for drafting code revisions and nomination of five members was made for a three-year term on the Committee on Research commencing in 1942. St. Paul was selected as the meeting place in June 1942.

Reports from Council Committees and Special Committees were received showing the current status of various activities.

In October the Council met at Washington, D. C. and established dues rates for 1942, heard a report on the progress of the 7th International Heating and Ventilating Exposition and voted on candidates for the F. Paul Anderson Award at the 48th Annual Meeting.

During the course of the year Life Memberships were granted to 12. Resignations of 60 members were accepted and cancellations of membership totaled 39. Death claimed 18 among whom were Life Members: J. J. Blackmore (also Charter Member), Alfred Kellogg, H. B. Gombers, E. B. Gordon, Jr., J. J. Nesbitt and W. A. Shorb.

Respectfully submitted,
THE COUNCIL.

The report of the Treasurer was presented by Mr. Blankin and emphasized the significant points in the Balance Sheet and the Budget Comparison prepared by the Society's Certified Public Accountant.

TUSA & LABELLA
CERTIFIED PUBLIC ACCOUNTANTS
52 WILLIAM ST., NEW YORK

January 16, 1942.

American Society of Heating and
Ventilating Engineers,
51 Madison Avenue,
New York, N. Y.

Gentlemen:

Pursuant to your request, we examined the books of account and records of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—New York, N. Y. and the related Funds for the year ended December 31, 1941 and submit herewith our report.

The audit covered a verification of the assets and liabilities as of the close of business December 31, 1941 and a review of the operating accounts for the year then ended. For the period audited the recorded cash receipts were traced into the depositories; the cancelled bank checks were inspected, compared with the cash records and supported by payment vouchers; also the dues income and interest income from savings accounts and securities was accounted for.

A Balance Sheet reflecting the financial condition of the Society as of the close of business December 31, 1941 is submitted herewith and your attention is directed to the following comments thereon:

CASH

Cash on Deposit and out for collection was verified by direct communication with the banks listed in the attached cash schedule and reconcilement of the balances reported to us with the amounts shown by the books of the Society.

The Cash on hand for Deposit and the Petty Cash was verified by count.

MARKETABLE SECURITIES

The securities, per the annexed schedule, were verified by direct communication with the Bankers Trust Company where same are deposited for safe-keeping. This asset has been included in The Balance Sheet at the cost of acquisition plus the accumulated and accrued interest earned thereon.

CERTIFICATE OF INDEBTEDNESS

During the current year the Society received a certificate of indebtedness No. 6 for the principal sum of $161.00 plus accumulated interest at 3%, for the period

from March 1, 1933 to December 31, 1940, amounting to $37.84 in settlement of their open account. In the future the 3% interest is payable only on the unpaid balance of the principal.

ACCOUNTS RECEIVABLE

A list of the membership dues receivable as of December 31, 1941 furnished to us by the management was checked to the individual ledger cards and found in agreement with the General Ledger Control. These unpaid dues were aged and may be summarized as follows:

Dues Invoiced during 1941	$ 6,944.00
Dues Invoiced during 1940	3,044.12
Dues Invoiced during Prior Years	3,583.59
Total	$13,571.71

Amounts due from Guide Advertisers and other debtors were verified by trial balance of the individual ledger accounts and found in agreement with the General Ledger Control.

The books of the Society reflect the following Reserves for Doubtful Accounts Receivable which we consider ample:

Reserve for unpaid dues	$7,501.54
Reserve for unpaid advertisements and sundry receivables	1,291.37
Total	$8,792.91

INVENTORIES

Items appearing under this caption are based upon quantities submitted by the management and computations made by us. The Transactions inventory priced at cost is scheduled below:

Volume	Year	Quantity	Price	Amount
-0-	Prior years	1750	$1.00	$1,750.00
41	1935	12	1.65	19.80
42	1936	24	1.65	39.60
43	1937	133	1.58	210.14
44	1938	146	1.58	230.68
45	1939	163	1.65	270.58
			Total	$2,520.80

PREPAID TRAVELING

As of December 31, 1941 the Society possessed Railroad Scrip Books which were acquired at a cost of $3,212.05. This asset was verified by actual count of the scrip books on deposit with the Executive Secretary which we found to aggregate the above sum.

PERMANENT ASSETS

Furniture, Fixtures and Library are shown herein at the book values without appraisal by us. We did, however, provide for depreciation of furniture and fixtures at the rate of ten (10%) per cent per annum.

ACCOUNTS PAYABLE

Amounts due creditors were determined by inspection of the unpaid bills found on file and examination of the record of cash disbursements made subsequently to December 31, 1941.

DEFERRED INCOME

Trial balances taken of the dues and miscellaneous accounts receivable disclosed prepayments of 1942 dues and Guide advertisements which we have shown in the subjoined balance sheet under deferred income.

RESERVE FOR TRANSACTIONS

There remain to be published TRANSACTIONS for 1940—Volume 46 and 1941 Volume 47. In providing reserves for these commitments we have retained the sum of $3,900.00 for Volume 46 and estimated the cost for Volume 47 at $3,700.00.

GENERAL FUND

An analysis of the General Fund showing the changes that occurred in this account during the calendar year 1941 follows:

BALANCE—JANUARY 1, 1941—PER FORMER REPORT		$84,892.16
ADDITIONS		
Excess of Income over expenses on Guide Operations for the Year ended December 31, 1941 from Guide Statement of Income and Expenses	$3,262.96	
Excess of Income over Expenses on Society activities for the year ended December 31, 1941 from Society Statement of Income and Expenses	1,823.49	
Exchange service plan accommodation Account closed out	106.76	
Accumulated Interest credited for the period from March 1, 1933 to December 31, 1940 in settlement of Account Payable to the Society	37.84	5,231.05
BALANCE—December 31, 1941—Per BALANCE SHEET		$90,123.21

SPECIFIC FUNDS

The cash and securities comprising the endowment and the F. Paul Anderson Funds were verified by direct communication. Also, the interest earned thereon was accounted for.

RESEARCH FUND

There is included also as part of this report a Balance Sheet showing the Financial position of the Research Laboratories as of the close of business December 31, 1941 as well as a statement of income and expenses and a Budget Comparison reflecting the operations from the calendar year 1941.

Respectfully submitted,
FRANK G. TUSA, (Signed)
Certified Public Accountant.

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

(December 31, 1941)

ASSETS

SOCIETY			
CASH			
On deposit	$53,175.91		
Out for collection	18.00		
On hand for deposit	52.14		
On hand	100.00	$53,346.05	
In closed bank		418.97	$ 53,765.02
RESERVE FUNDS			
Cash on deposit			11,353.04
INVESTMENTS (AT COST)			
Securities (Market Value $19,431.00)		18,074.06	
Add: Accumulated interest	1,200.00		
Accrued interest	37.50	1,237.50	19,311.56
CERTIFICATE OF INDEBTEDNESS		190.03	
Add: Accrued interest		2.28	192.31

ACCOUNTS RECEIVABLE			
Membership dues	$13,571.71		
Less: Reserve for doubtful	7,501.54	$ 6,070.17	
Advertisers and sundry debtors	29,989.35		
Less: Reserve for doubtful	1,291.37	28,697.98	$ 34,768.15
INVENTORIES			
Transactions—Copies	2,520.80		
Transactions—Paper	626.72	3,147.52	
Emblem and certificate frames		231.63	3,379.15
PREPAID TRAVELING			
Railroad scrip			3,212.05
PERMANENT			
Library		300.00	
Furniture and fixtures	2,650.96		
Less: Reserve for depreciation	946.57	1,704.39	2,004.39
			$127,985.67
SPECIFIC FUNDS			
ENDOWMENT FUND			
Cash on deposit	6,985.51		
Cash on hand for deposit	202.50	7,188.01	
Securities at cost (market value $16,236.40)	19,418.65		
Add: Accumulated interest	$250.00		
Accrued interest	78.96	328.96	19,747.61	26,935.62
F. PAUL ANDERSON FUND			
Cash on deposit			1,163.08
			$156,084.37

LIABILITIES AND GENERAL FUND

SOCIETY			
ACCOUNTS PAYABLE			$ 9,888.22
DUE RESEARCH FUND			
On dues as and when collected		$ 5,016.55	
For equipment purchased		187.12	5,203.67
ACCRUED ACCOUNTS			
Compensation—Secretary and staff			3,085.22
DEFERRED INCOME			
Members dues		9,126.72	
Proposed members dues		308.63	
Guide advertisements		2,650.00	12,085.35
RESERVE FOR PUBLICATIONS			
TRANSACTIONS—Volume 46		3,900.00	
TRANSACTIONS—Volume 47		3,700.00	7,600.00
TOTAL LIABILITIES			37,862.46
GENERAL FUND			
Society			90,123.21
			$127,985.67
SPECIFIC FUNDS			
ENDOWMENT FUND			
Principal		24,063.28	
Unexpended income		2,872.34	26,935.62
F. PAUL ANDERSON FUND			
Principal		1,000.00	
Unexpended income		163.08	1,163.08
			$156,084.37

Note "A": This Balance Sheet is subject to the comments contained in the letter attached to and forming a part of this report.

BUDGET COMPARISON

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

(For the Year Ended December 31, 1941)

	Actual	Budget Provision	Increases Decreases
INCOME:			
Initiation fees	$ 1,970.00	$ 1,200.00	$ 770.00
Membership dues	31,633.60	25,000.00	6,633.60
Editorial contract	15,999.96	16,000.00	.04
Profit—Emblems and frames	63.60	200.00	136.40
Profit—TRANSACTIONS, reprints, etc.	77.37	200.00	122.63
Interest from savings bank and securities	949.47	1,000.00	50.53
Sales of GUIDES	19,910.50	18,000.00	1,910.50
GUIDE advertising	27,046.72	24,000.00	3,046.72
International exposition	1,945.95	—0—	1,945.95
TOTAL	$99,597.17	$85,600.00	$13,997.17
EXPENSES:			
General Society Activities:			
Meetings	$ 2,503.00	$ 2,000.00	$ 503.00
Chapter allowances	1,000.00	1,200.00	200.00
Membership promotion	246.05	300.00	53.95
Speakers' bureau	1,408.45	1,500.00	91.55
HPAC Subscriptions	5,828.00	6,000.00	172.00
Postage	1,820.69	1,800.00	20.69
Rent and light	3,612.46	3,625.00	12.54
TRANSACTIONS	3,906.21	2,500.00	1,406.21
Codes	—0—	1,000.00	1,000.00
Salaries	18,156.00	18,000.00	156.00
General printing	491.97	500.00	8.03
Awards, membership certificates, etc.	231.54	200.00	31.54
President's fund	1,546.45	1,800.00	253.55
Secretary's travel	1,304.75	1,000.00	304.75
Council and chapter delegates travel	4,362.64	4,500.00	137.36
Bank charges	108.62	100.00	8.62
Multigraphing and address changes	525.75	400.00	125.75
Telephone	638.02	400.00	238.02
Telegrams	180.34	200.00	19.66
Professional services	575.00	600.00	25.00
General office expense	611.66	500.00	111.66
Office supplies	585.78	500.00	85.78
Furniture and fixtures	246.38	200.00	46.38
Reserve for additional compensation	3,085.22	3,250.00	164.78
Research appropriation	5,000.00	5,000.00	—0—
Special appropriation	100.00	245.00	145.00
Research booklets	890.04	750.00	140.04
	$58,965.02	$58,070.00	$ 895.02
GUIDE EXPENSES:			
Paper	$ 3,416.41	$ 3,000.00	$ 416.41
Printing and binding	14,491.56	13,300.00	1,191.56
Express and mailing	5,151.45	3,500.00	1,651.45
Engraving and art work	693.54	500.00	193.54
Advertising sales promotion	1,687.95	2,000.00	312.05
Copy sales promotion	3,144.92	3,000.00	144.92
Editorial and advertising salaries	5,199.95	6,000.00	800.05
Committee expenses	342.61	480.00	137.39
Yearbook	1,417.31	1,500.00	82.69
TOTAL GUIDE EXPENSES	$35,545.70	$33,280.00	$ 2,265.70
TOTALS	$94,510.72	$91,350.00	$ 3,160.72

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS RESEARCH FUND—NEW YORK, N. Y.

(December 31, 1941)

ASSETS

CASH			
ON DEPOSIT			
Treasurer's account—Bankers Trust Co	$1,850.10		
Secretary's account—Chase National Bank	2,571.76		
Director's account—Forbes National Bank	401.41		
Thrift account—Bank for Savings	3,252.16	$8,075.43	
ON HAND FOR DEPOSIT			
Treasurer's account—Bankers Trust Co		6,074.75	
ON HAND			
Petty cash—Pittsburgh, Pa		45.47	$14,195.65
ACCOUNTS RECEIVABLE			
AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS			187.12
CLAIM RECEIVABLE			
Navy Contract No. 66853			2,500.00
WORK IN PROGRESS			
Navy contract			3,354.94
PREPAID TRAVELING			91.15
PERMANENT			
Laboratory equipment			1.00
Total Research Funds			$20,329.86
RESEARCH ENDOWMENT FUND			
CASH ON DEPOSIT			
Bank for Savings		544.89	
Bank of U. S. in Liquidation		167.78	712.67
			$21,042.53

LIABILITIES AND FUNDS

DEFERRED INCOME			
PROJECTS			
Summer cooling	$1,859.45		
Glass study	1,088.38		
Corrosion	525.00		
Convectors	300.00	$3,772.83	
DUES—Members and Associates 40% of 1942 dues prepaid in 1941		5,330.00	$ 9,102.83
ACCRUED ACCOUNTS			
Professional service			100.00
RESEARCH FUND			11,127.03
Total Research Liabilities and Fund			$20,329.86
RESEARCH ENDOWMENT FUND			
Principal		600.00	
Unexpended income		112.67	712.67
			$21,042.53

Note "A": This Balance Sheet is subject to the comments contained in the letter attached to and forming a part of this report.

Note "B": The Research Fund as at December 31, 1941 had a contingent asset amounting to $5,016.55 arising from 40% of the Society's members' and associates' dues, payable upon collection by the latter.

BUDGET COMPARISON

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS RESEARCH FUND—NEW YORK, N. Y.

(For the Year Ended December 31, 1941)

	Actual	Budget Provision	Increases Decreases
INCOME:			
ASHVE dues	$17,987.22	$15,000.00	$ 2,987.22
ASHVE appropriation	5,000.00	5,000.00	....
Exposition	5,033.10	5,033.10	....
Interest	48.34	60.00	11.66
Contributions:			
General	2,836.00	2,500.00	336.00
Earmarked:			
Cooperative institutions	1,165.55	3,090.00	1,924.45
Navy contract	20,000.00	25,000.00	5,000.00
Radiation and comfort	935.00	1,400.00	465.00
Floor losses	1,000.00	1,000.00	....
Glass study	3.75	1,600.00	1,596.25
TOTALS	$54,008.96	$59,683.10	$ 5,674.14
EXPENSES:			
Cooperative Institutions	$12,775.30	$12,800.00	$ 24.70
Pittsburgh Laboratory:			
Navy contract	20,000.00	25,000.00	5,000.00
Radiation and comfort	3,495.15	3,400.00	95.15
Basement heat losses	994.84	1,000.00	5.16
Glass study	3.75	1,600.00	1,596.25
Air conditioning in industry	4,018.95	4,500.00	481.05
Duct friction	1,312.60	1,500.00	187.40
Cooling load	3,557.03	3,500.00	57.03
Heating load	3,096.04	3,000.00	96.04
Instruments and equipment	1,689.23	2,000.00	310.77
TOTALS	$50,942.89	$58,300.00	$ 7,357.11
Excess of income over expenses	3,066.07	1,383.10	1,682.97
TOTALS	$54,008.96	$59,683.10	$ 5,674.14

A. V. Hutchinson gave the report of the Secretary.

Secretary's Report

During the year 1941 the Secretary's office has carried on the administrative duties required by the Constitution and By-Laws of the Society. All details of the financial transactions will be given in the Treasurer's report. Those relating to the production of THE GUIDE 1942 will be outlined in the report of the Guide Publication Committee and other special committees will tell about the progress of their respective assignments.

The 1941 Annual and Semi-Annual Meetings were held at greater distances from headquarters than previous meetings of the Society and imposed a heavier load on the office staff in order to carry out the necessary preparations. Plans also had to be made for handling two expositions and the show held in San Francisco was considered an outstanding success. Unfortunately, the last minute cancellation of the Philadelphia Exposition required meeting program revisions and the necessity of issuing special publicity matter to members.

Due to the speeded-up program of Research, a greater number of papers have been developed this year than in a similar period in the past, so that the Publication Committee's work was greatly increased, but through the cooperation of Chairman P. Nicholls and his associates all essential requirements have been met in reviewing and printing the technical papers.

THE GUIDE has been enlarged somewhat and is being issued with a stiff binding in order to make a stronger and more useful volume. The mailing to members is being started during the week of January 26 and distribution should be completed by February 15.

XUM

Although some increase in membership was recorded a great many members have entered the military or naval services and many more have been engaged in emergency construction work so that the problem of keeping up with changes of addresses for members has been quite difficult in the past few months. Some trouble has been experienced in locating members whose mail was returned because they failed to notify the headquarters office of change of business or home address and left no forwarding address with the postal authorities.

The employment service was another phase of the work which has been in much greater volume this year than in former years. A great many placements have been made and there has been a constant demand from government agencies and firms engaged in defense work for experienced engineering personnel. In general, members of the profession have enjoyed a very prosperous year and this has been reflected in early dues collections.

A new method of providing speakers for Local Chapters was tried this year by the Local Chapters Committee and it was easier to handle and at the same time provided better service to the Chapters as they could depend upon definite dates for speakers. The Secretary was privileged to assist many of the Chapters in planning their meetings programs for the fall season and was able to visit 24 Chapter cities during the year.

Membership activities have been in good volume and during the year a new form of membership application was approved by the Council and is now in use. Each member received a copy recently with a new booklet describing the Aims and Activities of the Society. Earlier in the fall an outline of research activities was submitted to each member in a special booklet prepared by the Committee on Research.

Two volumes of TRANSACTIONS are now in preparation and Vol. 46 (1940) is being printed and will be ready for distribution within 60 days. This will be followed closely by Vol. 47. The Council requested that members be polled regarding their desire to have TRANSACTIONS and the return cards indicated that 75 per cent of the members desired transactions and filed their requests within the specified limit date.

It should be noted that the varied activities of the headquarters office were carried on without increase in staff during the past year and also that one half of the time of the Technical Secretary is devoted to the work of the Committee on Research. It is my desire to express appreciation for the splendid cooperation of the Officers and Committee Members who have served this year and to pay tribute to the fine work of the headquarters office staff.

Respectfully submitted,
A. V. HUTCHINSON, *Secretary.*

Prof. C.-E. A. Winslow was called upon by President Fleisher to present the report of the Committee on Research. Professor Winslow, Vice-Chairman of the Committee, gave a detailed report of the activities and outlined the problems now being studied at the Research Laboratory in Pittsburgh and at the Cooperating Institutions.

Committee on Research Annual Report 1941

One of the major accomplishments of the Committee on Research in 1941 was the preparation of a booklet, which was made available to the membership of the Society and other interested individuals. This publication contained for each of the 21 Technical Advisory Committees a comprehensive outline of carefully planned research investigations; regulations governing committee activities and a bibliography of research reports covering work emanating from the ASHVE Research Laboratory and cooperative institutions over a period of 22 years. The compilation of all this information was

intended to serve as a basis for coordinating the future research program of the Society.

A total of 16 papers and 4 committee reports were presented to the Society this year, as a result of research sponsored currently in 13 cooperating institutions and the Society Research Laboratory. This research program was conducted through the several Technical Advisory Committees, which include 261 active persons who were responsible for studying the requirements for, outlining plans for the carrying out of, and reviewing the results of research for which the Society expended over $55,000.

Another investigation was completed this year on the effect of radiation in a panel heated room. Other studies have dealt with determinations of heat loss and gain through roofs, walls and basement floors. Additional projects have covered friction heads in fittings, entrainment action of air streams, sound attenuation in duct systems, performance of stack heads, pressure loss in elbows, and the thermodynamic properties of moist air. Through the financial support of the Atlanta Chapter, one of the cooperative institutions was able to complete a study on the cooling of a structure with an attic fan.

The individual committee reports which follow give detailed information covering investigations, either conducted at the ASHVE Laboratory or in cooperating institutions.

Technical Advisory Committees

SENSATIONS OF COMFORT, *1*—Thomas Chester, *Chairman;* N. D. Adams, C. R. Bellamy, G. D. Fife, E. P. Heckel, Dr. W. J. McConnell, F. C. McIntosh, A. B. Newton, B. F. Raber, C. Tasker.

During the year a digest was prepared of eleven papers dealing with atmospheric conditions conducive to summer comfort in air conditioned enclosures, completing work which had been commenced during the previous year. This committee report entitled, Comfort with Summer Air Conditioning, will be presented at the 1942 Annual Meeting.

Due to the ASHVE Research Laboratory at Pittsburgh having been fully engaged in important government work and investigations under the auspices of other Technical Advisory Committees, the program prepared sometime ago relative to research for the purpose of eliminating obscurities surrounding important factors which affect sensations of comfort has not been initiated. It is hoped that progress will be made on this project when time and funds are available.

PHYSIOLOGICAL REACTIONS, *2*—C.-E. A. Winslow, *Chairman;* Dr. T. Bedford, Thomas Chester, Dr. E. F. DuBois, Dr. M. B. Ferderber, E. P. Heckel, John Howatt, Dr. R. W. Keeton, C. S. Leopold, André Missenard, Dr. R. R. Sayers, Charles Sheard, C. Tasker.

The chief work of the Committee as a whole has been the preparation of a report on the influence of atmospheric humidity on human health and comfort. This report is to be presented at the 1942 Annual Meeting of the Society. It summarizes the present status of knowledge with regard to the influences of atmospheric humidity, upon heat losses from the human body at high and low temperatures, respectively, and also with regard to the dehydrating effect of various atmospheric conditions upon the membranes of the nose and throat.

The University of Illinois, College of Medicine, has undertaken a study of the physiological adjustments, which follow the sudden entrance of an individual into an atmospheric environment of widely different heat properties from the one to which he was previously adjusted. At present the investigators are considering the sudden change from a comfortable to a hot moist environment (dry-bulb 99-100 F, wet-bulb

86-90 F). The period of adjustment is measured by the time required for the skin temperature of selected areas to become stable. Frequent records of electrocardiographic tracings have shown negligible changes in *T* waves. At present attention is concentrated on the venous pressure changes. On the sudden exposure to hot moist environments there is a distinct feeling of discomfort often accompanied by faintness and dizziness. In some sensitive individuals, with a sudden change in the position syncope results. Preceding syncope the arterial blood pressure falls rapidly and the heart beat is increased greatly. These changes have been attributed to an inadequate return of blood to the heart and to its accumulation in the venous channels. It was felt that if a satisfactory explanation could be obtained for this syncope, then the less marked changes attended by dizziness and discomfort could be more satisfactorily interpreted. At present healthy medical students are being used as subjects. It is planned to extend the observations to apparently healthy older individuals, and later to individuals with well compensated heart disease.

AIR CONDITIONING IN INDUSTRY, *3*—W. L. Fleisher, *Chairman;* Dr. Leonard Greenburg, H. P. Greenwald, W. C. L. Hemeon, A. M. Kinney, J. W. Kreuttner, L. L. Lewis, H. B. Matzen, Dr. W. J. McConnell, Dr. C. P. McCord, P. A. McKittrick, Dr. R. R. Sayers, Charles Sheard, C. Tasker, R. M. Watt, Jr., H. E. Ziel.

The principal work accomplished this year has been rather exhaustive tests in spot cooling with different temperatures, humidities and velocities, to determine whether, by introducing air at conditions below the skin temperature, relief could be afforded workmen who are in such intensely hot industries, that the conditioning of the entire enclosure is not feasible.

Although actual tests were not conducted to determine the reactions of women workers engaged in industrial activities, the relative effect of the same ambient condition on men and women has been observed to a certain extent in other investigations conducted at the ASHVE Research Laboratory this year from which conclusions may be drawn.

Some progress has been made on an investigation to determine critical conditions of the atmosphere at low relative humidities, with special emphasis on the fatigue of leucocytes in connection with high temperatures and low humidities.

REMOVAL OF ATMOSPHERIC IMPURITIES, *4*—Dr. Leonard Greenburg, *Chairman;* J. J. Burke, J. M. DallaValle, R. S. Dill, Theodore Hatch, L. R. Koller, F. H. Munkelt, H. C. Murphy, G. W. Penney, Dr. E. B. Phelps, F. B. Rowley, C. Tasker, W. O. Vedder, J. H. Waggoner, R. P. Warren, W. F. Wells, Dr. Renée Wiener.

The program of this committee entails the preparation of a comprehensive outline of work already done in the field, collecting known data and summarizing it for interpretation by the engineer, and indicating gaps of knowledge for which subsequent research may be conducted.

The work is divided into three divisions formulated to deal with the special aspects of the problem, which include mechanical, chemical and bactericidal methods of purification.

1. *Mechanical Purification.* The collection of factual data covering various types of dusts encountered in industrial operations at the point of production. Samples to be submitted to careful microscopic particle size determinations and separated into various classifications as follows: (*A*) Under 20 microns; (*B*) Under 10 microns; (*C*) Under 1 micron.

It is planned to extend this work until sufficient data have been obtained to represent a satisfactory picture of the usual type of industrial dust producing processes and filtration devices.

2. *Chemical Purification.* Such processes involving adsorption, decomposition, oxidation and reaction are considered herein as chemical methods of purification. The use of chemical processes for air purification is limited properly to gas and vapor impurities. However, chemical means may be used to prevent contamination of air by gases from solid or liquid matter, such as chlorination or liming of fetid or decaying matter and it may be that some method of this type will require the atten-

tion of the committee in the future. Four methods of chemical air purification are being studied: Adsorption by activated carbon, oxidation by ozone, washing with spray water containing such chemicals as chlorine or caustics in solution, and vaporization, or volatilization, or atomization of chemicals in air.

3. *Bactericidal Purification.* The development of literature and the clinical examination of experimental data involving the utilization of ultra violet radiation sources.

RADIATION AND COMFORT, 5—J. C. Fitts, *Chairman;* A. H. Barker, L. M. K. Boelter, R. E. Daly, J. B. Fullman, E. R. Gurney, L. N. Hunter, A. P. Kratz, C. S. Leopold, D. W. Nelson, W. J. Olvany, G. W. Penney, W. R. Rhoton, C. J. Stermer, C.-E. A. Winslow.

This committee was formed in 1938 for the purpose of considering all problems related to radiation as it affects human comfort in winter and summer. From an outline of basic researches prepared, the committee initially decided to concentrate on a laboratory investigation to determine the primary sense reactions of relative comfort derived from rooms heated by convection and direct radiation, and with normal window arrangements.

During the summer of 1939 the necessary building was erected at the ASHVE Research Laboratory in Pittsburgh, and throughout the winter tests were conducted, the results of which were submitted in a paper at the 1941 Annual Meeting.

Subsequently, one of the test rooms in the building was equipped with ceiling panels constructed of sinuous coils through which hot water was circulated to provide all of the heating requirements. Tests conducted through the winter of 1940-41 resulted in a second paper dealing with radiation as a factor in the feeling of warmth in convection, radiator and panel heated rooms, which is to be presented at the 1942 Annual Meeting.

In addition, the committee has been responsible for directing a cooperative research project at the University of California, which is intended to develop a more rational basis for the design of panel heating and cooling systems. A progress report covering this work is also being presented at the 1942 Annual Meeting.

INSTRUMENTS, 6—D. W. Nelson, *Chairman;* F. R. Bichowsky, L. M. K. Boelter, R. S. Dill, A. P. Gagge, J. A. Goff, A. E. Hershey, F. C. McIntosh, F. W. Reichelderfer, G. L. Tuve, C. P. Yaglou.

Progress has been made this year in establishing standards for measuring the physical properties of the thermal environment: namely, relative humidity, ambient air temperature, air movement and mean radiant wall temperature. The committee has agreed on the measurement of these items and expects to have a report available in the near future.

Much work has been accomplished on the measurement of radiant temperatures. While recognizing that investigations in progress may alter the situation, the committee is prepared to agree on the blackened globe thermometer in 8 in. size for measuring the mean radiant temperature. Its use is based on the principle that radiant energy is exchanged between its spherical surface and all surfaces it *sees* and it loses heat by convection at a rate dependent on this energy reception, the ambient temperature and the air movement at its location. A more direct method, the use of thermopile, measures radiant temperatures of each surface directly but is tedious to use because of the many readings required and the necessary integration to obtain the answer related to an object such as the human body.

The means of measuring low velocity air currents as encountered in heated and cooled rooms is receiving considerable attention at the present, with most emphasis being placed on some form of thermo anemometer, such as the hot wire anemometer. Several methods of calibration are being attempted in the various laboratories doing society cooperative research in the field of air distribution, radiant heating and other projects involving the thermal environment. The wind tunnel seems to be first in favor with the whirling arm and the gasometer also being investigated. The problem appears to be to obtain a known linear air motion in the test section for calibration and relate that to the non-linear or miscellaneous free motion in the room air to be measured.

WEATHER DESIGN CONDITIONS, *7*—T. H. Urdahl, *Chairman;* J. C. Albright, H. S. Birkett, P. D. Close, John Everetts, Jr., C. M. Humphreys, O. A. Kinzer, H. H. Koster, J. W. O'Neill, F. W. Reichelderfer.

During the past year cooperative work has been initiated in Washington, D. C. to determine temperature differences existing between business and suburban sections. Several recording stations were selected at representative points in the city, taking into consideration the geological characteristics of the locality. Simultaneous readings obtained with the official Weather Bureau record were observed to establish a percentage of variations for the various sections of the city selected for observation. The conduct of this work in Washington will enable the development of a procedure which may be applied to other localities, in order to obtain these data for each representative city. In this work the cooperation of local utilities, colleges and universities, and especially local Society Chapters has been earnestly sought to provide the personnel required for making the observations.

RADIATION WITH GRAVITY AIR CIRCULATION, *8*—M. K. Fahnestock, *Chairman;* R. E. Daly, R. S. Dill, A. G. Dixon, H. F. Hutzel, J. P. Magos, J. W. McElgin, J. F. McIntire, T. A. Novotney, W. A. Rowe.

The work of this committee during the past year at the University of Illinois and in other cooperating laboratories was a continuation of a program started in 1940 which included:

1. Studies to determine the influence of a cold-wall test room and a warm-wall test booth on the performance of radiators and convectors, and the development of test methods which would give similar output ratings with each type of test equipment. Two direct cast-iron radiators, one cast-iron convector, and one non-ferrous convector were tested in the cold room and in the warm-wall booth at the University of Illinois.

2. A study of tests conducted with the same radiators and convectors in a number of different laboratories for the purpose of improving test methods and developing test equipment which would make possible more uniformity in test results. This work involved tests with two cast-iron radiators, two cast-iron convectors, and one non-ferrous convector in three cold rooms, three cold-wall test rooms, and three warm-wall test booths. The tests have been completed in three cold rooms, in one cold-wall test room, and in two warm-wall test booths located in different laboratories.

3. A study of the effect of the ambient air temperature on the performance of radiators and convectors tested in rooms surrounded by cold spaces. Tests have been completed with two cast-iron radiators, two non-ferrous convectors, and one cast-iron convector in the cold room at the University of Illinois, with temperatures in the spaces surrounding the cold room ranging from — 5 to + 12 F.

A second warm-wall test booth and auxiliary equipment are being provided for a study of the performance of convectors and radiators when used with hot water. It is proposed to conduct tests with one cast-iron convector, one non-ferrous convector, and one small-tube cast-iron radiator, and to determine the fluid pressure loss and the heat output for each unit under each operating condition. The operating conditions will include mean water temperatures of 140, 170, 200, and 210 F, and water temperature drops of 5, 10, and 15 F for each mean water temperature. Tests will be made with ½ in. piping connections and with connections the same size as the tapping in the heaters. Both sizes of connections will be used with end supply and return and with underneath supply and return piping arrangements.

HEAT TRANSFER OF FINNED TUBES WITH FORCED AIR CIRCULATION, *9*—W. E. Heibel, *Chairman;* C. M. Ashley, William Goodman, H. F. Hutzel, Ferdinand Jehle, S. F. Nicoll, R. H. Norris, L. P. Saunders, R. J. Tenkonohy, G. L. Tuve, C. F. Wood.

One function of this committee has been to develop fundamental data so that more accurate coefficients could be established for testing and calculating sizes of finned tubes. Studies outlined during the current year include:

Determination of factors involved in the transfer of heat through water films existing on the external surface of finned tubes of various types.

Determination of minimum refrigerant temperature for operation of finned tubes above the frost line on the air side of the coil.

A study of the performance of finned tubes under operation with volatile and non-volatile refrigerants.

A study of the effect of temperature, both on the air side and on the heating or cooling medium side of the surface.

Standardization in terminology, symbols and arrangement of data should be established.

Standardized types of instruments should be selected, including: temperature, electrical, humidity, pressure, air flow, refrigerant, time and weight measuring devices.

COOLING LOAD IN SUMMER AIR CONDITIONING, *10*—C. M. Ashley, *Chairman;* John Everetts, Jr., E. H. Hyde, C. S. Leopold, C. O. Mackey, R. M. Stikeleather, J. H. Walker, W. E. Zieber.

A report giving the twenty-four hour heat gain rates for 11 walls and 18 roofs, having wide variation in resistance to heat flow and capacity, will be presented at the 1942 Annual Meeting, as a contribution of this committee. The data presented in this paper are considered important as an initial contribution towards a more accurate method of determining the cooling load of a structure. Further study of the data is necessary to determine how the observed heat flows compare with the theory relating heat flow to the orientation, heat capacity, transmittance coefficient and absorptivity of solar radiation by the outside surface of the structure.

The committee program also includes problems relating to infiltration through open, swinging and revolving doors, as well as the important question of establishing more accurate heat gain data from gas, electric and steam appliances.

SUMMER AIR CONDITIONING FOR RESIDENCES, *11*—M. K. Fahnestock, *Chairman;* C. F. Boester, E. A. Brandt, John Everetts, Jr., H. F. Hutzel, E. D. Milener, K. W. Miller, F. G. Sedgwick, J. H. Walker.

While no active research work was done under the auspices of this committee during the summer of 1941, a comprehensive program of future investigation was outlined. The erection of a new research home at the University of Illinois makes available a completely equipped residence varying in size and type of construction from the home used for previous investigations.

The new residence has six medium sized rooms. The brick-veneer exterior walls and the second floor ceiling are insulated with full thickness insulation. In addition to repeating some of the work which has been conducted in the eight room wood frame house, work in the new house might include:

1. Studies to determine the practical effectiveness of various window appurtenances in reducing the cooling load.

2. Studies to determine the practical effectiveness of storm windows, double glazed windows, and windows of special glass on the cooling load.

3. Studies to determine the difference in the comfort of a small home, first, when the attic is kept closed continuously, and second, when it is ventilated continuously by gravity through built-in louvers or windows.

4. Studies to determine the capacity of attic fans to be used to obtain reasonable and practical benefits from the circulation of outdoor air at night.

5. Studies to determine the feasibility and practicability of using one or more mechanical unit room coolers to cool a small residence.

AIR DISTRIBUTION AND AIR FRICTION, *12*—J. H. Van Alsburg, *Chairman;* S. H. Downs, A. E. Hershey, R. D. Madison, L. G. Miller, D. W. Nelson, C. H. Randolph, D. J. Stewart, M. C. Stuart, Ernest Szekely, R. J. Tenkonohy, G. L. Tuve.

Five years of continuous activity by this committee on problems related to air distribution and air friction has reached a point during the past year where investigations were being conducted not only at the ASHVE Research Laboratory, but also in five cooperating institutions.

Studies on the frictional resistance to the flow of air in round pipes has been completed at the Society Laboratory and results of tests on square and rectangular ducts is planned for presentation at the 1942 Semi-Annual Meeting. A paper covering the pressure losses in round elbows as the result of cooperative research with Lehigh

University, will be presented at the 1942 Annual Meeting. Future investigations at this institution have been planned to develop data on the pressure losses, resulting from turning vanes placed in elbows. The resistance of duct transitions and Y fittings is the subject of an investigation being conducted at Michigan State College.

In the field of air distribution a paper on the performance of stack heads equipped with grilles will also be presented at the 1942 Annual Meeting, covering work recently completed at the University of Wisconsin. As a result of cooperative research at the Case School of Applied Science, another paper on the entrainment and jet-pump action of air streams will be available for the 1942 Annual Meeting.

Another project on air distribution, involving the effect of inlet and exhaust opening location on the motion of air within a room, is being actively investigated at the University of Illinois. It is expected that results from this study will be released for publication in the near future.

HEAT REQUIREMENTS OF BUILDINGS, *13*—P. D. Close, *Chairman;* E. K. Campbell, J. F. Collins, Jr., E. F. Dawson, W. H. Driscoll, H. M. Hart, E. C. Lloyd, H. H. Mather, C. H. Pesterfield, F. B. Rowley, R. K. Thulman, J. H. Walker.

A study of basement heat losses has been in progress at the ASHVE Research Laboratory in Pittsburgh during the past year and as a result of this study a paper entitled, Heat Loss through Basement Walls and Floors, will be presented at the 1942 Annual Meeting. These tests were conducted in a simulated basement 15 x 15 x 8 feet high, built on the premises of the Bureau of Mines in Pittsburgh. This report should furnish a basis for revision of the GUIDE data on calculation of basement heat losses and also add valuable data on the prevention of condensation on basement walls. However, it appears from this research report that a further study of the subject is desirable.

Two other papers dealing with heat requirements of buildings have also been prepared under the sponsorship of this committee. One of these entitled, An Analysis of Factors Influencing Building Heat Losses, was presented at the 1941 Semi-Annual Meeting of the Society. The other paper entitled, Attic Temperatures, Ventilation and Heat Losses, will be published in the near future and is intended to throw some light on a phase of the heat loss problem to which little consideration has been given until recently.

At the request of the Society's Joint Committee to Study Uniform Methods of Heat Loss Calculation, consideration is being given by this committee to the preparation of a condensed manual for heat loss calculations based on the GUIDE procedure, but is not intended to replace the more accurate GUIDE method.

AIR CONDITIONING REQUIREMENTS OF GLASS, *14*—M. L. Carr, *Chairman;* C. M. Ashley, L. T. Avery, F. L. Bishop, D. A. Bridges, W. A. Danielson, H. C. Dickinson, J. D. Edwards, J. E. Frazier, S. O. Hall, E. H. Hobbie, C. L. Kribs, Jr., Axel Marin, R. A. Miller, F. W. Parkinson, W. C. Randall, L. T. Sherwood, J. T. Staples, G. B. Watkins, F. C. Weinert.

No meetings were held by this committee in 1941, but one proposed activity was considered by correspondence early in the year and it was later decided to drop the project until adequate laboratory facilities could be arranged.

During the year a paper was published resulting from the work of this committee, which included a review of glass transmittance coefficients. The results of the several studies cited in this paper indicated that the transmittance coefficients for glass varied over a wide range. It was found that the reported variations could not be satisfactorily correlated with peculiarities in installation, nor could they be with such test conditions as the air velocity over the test surface, the method of heat input, the size of the panel tested or the location and the method of temperature measurement. It was felt that these reported studies would serve as a basis for outlining a future and more comprehensive study of the entire project.

A suggestion has been received to continue the study of heat transmission through flat glass and single and double glazing in the near future.

INSULATION, *15*—R. T. Miller, *Chairman;* E. A. Allcut, R. E. Backstrom, Wharton Clay, R. E. Daly, H. C. Dickinson, J. D. Edwards, W. V. Hukill, E. C. Lloyd, Paul McDermott, E. W. McMullen, W. T. Miller, E. R. Queer, F. B. Rowley, W. S. Steele, J. H. Waggoner, G. B. Wilkes.

Plans have been formulated for investigating three projects which are of special interest to this committee. One deals with additional studies to determine the conductances of various air spaces bounded with reflective materials. Another project involves a careful investigation covering the conductivity of materials subjected to variable heat flow. It is also felt that additional research is desirable to determine the heat capacity of various building and insulating materials.

SOUND CONTROL, *16*—J. S. Parkinson, *Chairman;* C. M. Ashley, A. L. Kimball, V. O. Knudsen, R. D. Madison, R. F. Norris, C. H. Randolph, A. E. Stacey, Jr., A. G. Sutcliffe, Thomas A. Walters, R. M. Watt, Jr.

The research program of this committee has been designed to procure fundamental data in order to establish background noise levels encountered in air conditioned spaces.

Cooperative work with Rensselaer Polytechnic Institute was continued this year to obtain additional information for predicting sound attenuation through a duct system. A paper covering the application of a method for determining sound absorption in a duct system developed at this institution will be presented at the 1942 Annual Meeting.

A tentative testing procedure has been developed by the committee for measuring the amount of noise generated by equipment. The committee is encouraging individual experimentation to check the principles outlined in this standard, in order that necessary revisions to the procedure may be subsequently considered.

COOLING TOWERS, EVAPORATIVE CONDENSERS AND SPRAY PONDS, *17*—B. M. Woods, *Chairman;* W. W. Cockins, S. C. Coey, E. H. Kendall, S. R. Lewis, F. C. McIntosh, H. B. Nottage, J. F. Park, E. T. Selig, Jr., E. W. Simons, E. H. Taze.

The work on cooling towers and spray ponds has progressed along the following course since the previous report.

1. Tests on a spray tower, with several different types of nozzles (3) were finished.
2. Further work is being done, especially with the purpose of evaluating nozzle performance.
3. Test results on two types of nozzles in a spray pond have been evaluated.
4. The performance of spray towers and ponds has been presented on a unit volume conductance basis.
5. A method of measuring unit mass conductances of single elements has been proposed in collaboration with others.

To date five papers have been released for publication in the Society JOURNAL and other technical literature covering the work of this committee. The staff at the University of California contributed a section on cooling tower design for inclusion in the 1942 edition of the GUIDE.

PSYCHROMETRY, *18*—F. R. Bichowsky, *Chairman;* D. B. Brooks, W. H. Carrier, H. C. Dickinson, A. W. Gauger, J. A. Goff, William Goodman, A. M. Greene, Jr., L. P. Harrison, F. G. Keyes, A. P. Kratz, D. M. Little, Axel Marin, D. W. Nelson, W. M. Sawdon, F. O. Urban.

The first progress report sponsored by this committee was presented at the 1941 Semi-Annual Meeting covering cooperative work conducted at the University of Pennsylvania. This paper gave preliminary results on the experimental measurement of the interaction constant for moist air at a temperature of 15 C.

Owing to the desirability of making some improvements in the apparatus before attempting to obtain final results covering as wide a range of temperature as possible, no additional data could be presented at the present time. However, a paper covering recalculations of the vapor pressure of ice, which is a part of the ultimate objective of the committee in formulating more accurate thermodynamic properties of moist air, will be presented at the 1942 Annual Meeting.

XUM

CORROSION, *19*—A. R. Mumford, *Chairman;* H. E. Adams, N. D. Adams, J. F. Barkley, L. F. Collins, W. H. Driscoll, T. J. Finnegan, W. Z. Friend, R. R. Seeber, E. T. Selig, Jr., F. N. Speller, C. M. Sterne, J. H. Walker.

Plans have been formulated for a long range program of investigation, and financial support has been obtained for the first phase of this program, which will cover an investigation of the solubility of carbon dioxide in otherwise pure condensate at low partial pressures and at condensate temperatures.

The committee has been considering a preliminary report of the cooperative work at the Michigan College of Mining and Technology, where it is planned to continue this investigation without expense to the Society. A paper covering corrosion of several alloys in air washers of air conditioning systems was prepared by a member of the committee for presentation at the 1942 Annual Meeting of the Society.

In line with its long range program, the committee has initiated and is now considering a paper presented from industry on the present non-technical status of counter-corrosion agents, such as are met with in the group of organic amines. It is planned that this paper will be available for consideration by the Society in the near future.

FLOW OF FLUIDS THROUGH PIPES AND FITTINGS, *20*—S. R. Lewis, *Chairman;* L. A. Cherry, G. C. Davis, T. M. Dugan, Earle W. Gray, R. T. Kern, H. A. Lockhart, Axel Marin, R. F. Taylor, E. L. Weber.

The major objective of this committee is to determine proper allowances for resistances to flow of fluids of various kinds through pipes, valves, fittings, heat receivers and transmitters.

The activities of the committee have been confined largely to a cooperative research project being investigated at the Agricultural and Mechanical College of Texas. Special earmarked funds were solicited to cover the expense of testing a large section of six-inch pipe with valves, fittings and sections of both welded and screwed methods of assembly.

The first phase of this study has been completed and results covering a comparison of the friction heads in screwed and welded elbows are to be presented at the 1942 Annual Meeting. Studies to determine the influence of temperature on the loss of head and the friction loss resulting from corrosion of the inner pipe surface are now under investigation.

FUELS, *21*—R. A. Sherman, *Chairman;* R. M. Conner, R. S. Dill, A. C. Fieldner, L. N. Hunter, S. Konzo, W. M. Myler, Jr., H. J. Rose, C. E. Shaffer, T. H. Smoot, R. K. Thulman, E. C. Webb.

This committee was appointed in the summer of 1941 and has held no meetings but has organized itself for future activities by correspondence. The scope of the committee has been defined as:

The committee is concerned with the production of heat for comfort heating and for service hot water as in residences, apartments and office buildings. This includes the characteristics of fuels, the mechanism of the combustion of fuels, the design of equipment for the combustion of fuels, the design of boilers, furnaces, and space heaters, the design of automatic controls for such equipment, and the equipment by which the waste products, ash and flue gases, are removed from the heating equipment. Its functions are the encouragement of dissemination of existing knowledge in these fields by papers presented before local and national meetings of the Society and publication in the JOURNAL, and the initiation, support, and supervision of research to obtain information now lacking.

Suggestions have been received for a number of papers for technical sessions of the Society and for problems on which research projects are needed. At an early meeting of the committee, it is expected that one or two projects will be selected for sponsorship by the committee and steps taken to initiate the work.

INSTITUTIONS COOPERATING WITH COMMITTEE ON RESEARCH

Agricultural & Mechanical College of Texas: Frictional Flow of Water in Pipes and Elbows; Heating Requirements of Buildings. *University of California:* Perform-

ance of Cooling Towers; Radiant Heating and Cooling. *Case School of Applied Science:* Air Distribution in Air Conditioned Spaces. *Georgia School of Technology:* Cooling of a Structure with Attic Fans. *University of Illinois, Engineering:* Radiator and Convector Studies; Residence Summer Air Conditioning; Air Distribution Outlets. *University of Illinois, Medical School:* Human Responses to Physiological Reactions. *Lehigh University:* Frictional Resistance to the Flow of Air Through Elbows. *Michigan College of Mining & Technology:* Corrosion in Steam Heating Systems. *Michigan State College:* Air Flow in Duct Transitions. *University of Minnesota:* Methods of Rating Air Cleaning Devices. *University of Pennsylvania:* Measuring Departures from Dalton's Law of Air-Water Vapor Mixtures. *Rensselaer Polytechnic Institute:* Sound Transmission in Ducts. *University of Wisconsin:* Effect of Entering Air Temperature and Velocity on Distribution of Air in Enclosed Spaces.

RESEARCH PAPERS—1941

Cooperative Institutions

1. The Interaction Constant for Moist Air, by John A. Goff and A. C. Bates (Pennsylvania) (ASHVE TRANSACTIONS, Vol. 47, 1941, No. 1186).

2. Economical Air Velocities for Mechanical Air Filtration, by Frank B. Rowley and Richard C. Jordan (Minnesota) (ASHVE TRANSACTIONS, Vol. 47, 1941, No. 1187).

3. Performance Characteristics of a Forced Draft, Counterflow Spray Cooling Tower, by Howard H. Niederman, E. D. Howe, John P. Longwell, Ralph A. Seban and L. M. K. Boelter (California) (ASHVE TRANSACTIONS, Vol. 47, 1941, No. 1189).

4. Pressure Loss Caused by Elbows in 8-inch Round Ventilating Duct, by M. C. Stuart, C. F. Warner and W. C. Roberts (Lehigh) (see p. 335).

5. Entrainment and Jet-Pump Action of Air Streams, by G. L. Tuve, G. B. Priester and D. K. Wright, Jr. (Case) (see p. 241).

6. The Performance of Stack Heads Equipped with Grilles, by D. W. Nelson, D. H. Lamb and G. E. Smedberg (Wisconsin) (see p. 279).

7. Determining Sound Attenuation in Air Conditioning Systems, by D. A. Wilbur and R. F. Simons (Rensselaer) (see p. 267).

8. The Effect of Attic Fan Operation on the Cooling of a Structure, by W. A. Hinton and Albert F. Poor (Georgia Tech.) (see p. 145).

9. Panel Heating and Cooling Performance Studies, by B. F. Raber and F. W. Hutchinson (California) (see p. 35).

10. A Comparative Study of the Friction Heads in Screwed and Welded Elbows, by F. E. Giesecke and J. S. Hopper (Texas) (see p. 201).

11. The Vapor Pressure of Ice from 32 to −280 F, by John A. Goff (Pennsylvania) (see p. 299).

ASHVE Laboratory

12. Local Cooling of Workers in Hot Industry, by F. C. Houghten, M. B. Ferderber and Carl Gutberlet (ASHVE TRANSACTIONS, Vol. 47, 1941, No. 1188).

13. A Review of Glass Transmittance Coefficients, by F. C. Houghten, M. L. Carr and David Shore (November 1941, JOURNAL).

14. Radiation as a Factor in the Feeling of Warmth in Convection, Radiator and Panel Heated Rooms, by F. C. Houghten, Carl Gutberlet and E. C. Hach (see p. 55).

15. Heat Gain Through Walls and Roofs as Affected by Solar Radiation, by F. C. Houghten, E. C. Hach, S. I. Taimuty and Carl Gutberlet (see p. 91).

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16. Heat Loss Through Basement Walls and Floors, by F. C. Houghten, S. I. Taimuty, Carl Gutberlet and C. L. Brown (see p. 369).

Technical Advisory Committees

17. Comfort with Summer Air Conditioning, by Technical Advisory Committee on Sensations of Comfort, Thomas Chester, *Chairman* (see p. 107).

18. The Physiological Influence of Atmospheric Humidity, by the Technical Advisory Committee on Physiological Reactions, C.-E. A. Winslow, *Chairman* (see p. 317).

19. Corrosion Tests in a Water-Recirculating Air Conditioning System, by W. Z. Friend (Sponsored by Technical Advisory Committee on Corrosion) (see p. 233).

20. Measurement of the Physical Properties of the Thermal Environment, by Technical Advisory Committee on Instruments, D. W. Nelson, *Chairman* (Available mimeographed form).

CONTRIBUTORS TO RESEARCH

Contributions in support of the research work of the Committee on Research are hereby acknowledged:

Financial Contributions

Aerofin Corp.; *Air Conditioning Association of Cleveland;* American Air Filter Co., Inc.; American Radiator & Standard Sanitary Corp.; Anemostat Corporation of America; Atlanta Chapter of ASHVE; Auer Register Co.; Automatic Burner Corp.; Barber-Colman Co.; Bell & Gossett Co.; Buffalo Forge Co.; A. M. Byers Co.; Carrier Corp.; Consolidated Gas Electric Light & Power Company of Baltimore; Crane Co.; Detroit Edison Co.; Thomas M. Dugan; Frick Co., Inc.; G & O Mfg. Co.; General Electric Co.; Georgia Power Co.; Hart & Cooley Mfg. Co.; *Heating, Piping & Air Conditioning Contractors National Association;* Hoffman Specialty Co.; Ilg Electric Ventilating Co.; Illinois Testing Laboratories, Inc.

Insulation Board Institute; Johns-Manville Corp.; Johnson Service Co.; Kinetic Chemicals, Inc.; Minneapolis-Honeywell Regulator Co.; Mueller Brass Co.; Narowetz Heating & Ventilating Co.; Nash Engineering Co.; National Radiator Co.; New York Steam Corp.; Owens-Illinois Glass Co.; Permutit Co.; Philadelphia Electric Co.; Pittsburgh Corning Corp.; Sarco Co., Inc.; Staynew Filter Corp.; Surface Combustion Corp.; Taylor Instrument Companies; Trane Co.; Union Electric Company of Missouri; Utica Radiator Corp.; Warren Webster & Co.; Weil-McLain Co.; Westinghouse Electric & Mfg. Co.; L. J. Wing Mfg. Co.; Worthington Pump and Machinery Corp.; York Ice Machinery Corp.

Donors of Equipment or Material

American Radiator & Standard Sanitary Corp.; Barber-Colman Co.; Bell & Gossett Co.; Burnham Boiler Corp.; A. M. Byers Co.; Detroit Lubricator Co.; Eagle-Picher Lead Co.; Excelsior Steel Furnace Co.; Exposition Cotton Mills; Julien P. Friez & Sons; Furblo Co.; Gloninger Brick Co.; Hart & Cooley Mfg. Co.; Independent Register Co.; International Heater Co.; Johnson Service Co.; Koppers Co.; Lamneck Products, Inc.: Lau Blower Co.; Modine Mfg. Co.; Owens-Corning Fiberglas Corp.; *Structural Clay Products Institute;* Tuttle & Bailey, Inc.; Waterloo Register Co.

Periodicals

Air Conditioning and Oil Heat; American Artisan; Canadian Engineer; Domestic Engineering; The Engineer (London); *Heating, Piping and Air Conditioning; Heating and Ventilating; Ice and Refrigeration; Journal of Institution of Heating and Ventilating Engineers* (London); *Plumbing and Heating Journal.*

In the absence of Prof. C. M. Humphreys, C. S. Leopold, Member of the Guide Publication Committee, presented the report.

Report of the Guide Publication Committee

Work on the 20th edition of the HEATING, VENTILATING, AIR CONDITIONING GUIDE 1942 was started when the Guide Publication Committee held its organization meeting on March 8, 1941. Two additional meetings were held during the spring and summer. In the preparation of this GUIDE, the Committee has endeavored to present the latest available authoritative information on the various subjects covered in the 47 Chapters. In accordance with the policy established by its founders, approximately one-half of the GUIDE chapters have been reviewed. Of these, nine were completely rewritten and minor revisions were made to nine others.

A new chapter entitled Fundamentals of Heat Transfer has been added which includes the basic equations for conduction, convection, and radiation. A detailed solution is given for a problem involving all three mechanisms of heat transfer. Except for the insertion of this chapter, there has been no change in the order of subject presentation.

Some revisions have been made in the text of the chapter on Thermodynamics of Air and Water Mixtures. The Mollier Diagram for Moist Air, which first appeared in the GUIDE 1941 has been redrawn and a new Volume Diagram for Moist Air has been added.

The chapter on Central Systems for Comfort Air Conditioning has been completely rewritten, and the revision of the chapters on Air Distribution, Air Duct Design, Sound Control, Fans provides new data on the design of air handling systems. The new air friction chart developed by the ASHVE Research Laboratory is included in the Air Duct Design chapter. The chapter on Radiant Heating has undergone major revisions, and now contains the essential data necessary for the design of radiant heating systems.

In addition to the chapters mentioned above, the following were also reviewed and revised: Physiological Principles, Combustion and Fuels, Automatic Fuel Burning Equipment, Radiators and Convectors, Pipe, Fittings, Welding, Heat Transfer Surface Coils, Spray Equipment, Air Pollution, Air Cleaning Devices, Natural Ventilation, and Water Supply Piping and Water Heating.

The Technical Data Section of the 1942 GUIDE contains 856 pages, an increase of 48 pages over the previous issue. This increase was made absolutely necessary by the addition of new and important material.

The Catalog Data Section contains valuable and concise information on many types of equipment. It has been carefully arranged in logical sub-divisions to facilitate its use.

The 1942 GUIDE is the result of the unselfish work of many individuals over a period of 20 years. In recognition of this fact, the Guide Publication Committee wishes to thank the many contributors to the earlier editions of the book. We also thank the many GUIDE users who have offered helpful suggestions, and those who assisted the Committee in reviewing and editing the GUIDE material. Special thanks are due those who have contributed the new material which appears in the 1942 GUIDE. Their names are included in the Preface.

The Committee also wishes to acknowledge the excellent work of John James, Technical Secretary of the Society. His expert guidance and judgment were invaluable.

As Chairman, I wish to thank the other members of the Committee who so willingly devoted their time and energy to the task of producing a new GUIDE.

In closing, the Committee is glad to announce that many GUIDES are already completed and are now being distributed. We hope that the 1942 GUIDE may prove as useful as its predecessors.

Respectfully submitted,
CLARK M. HUMPHREYS, *Chairman.*

Joe Wheeler, Jr., Chairman of the Board of Tellers, gave the result of election of Officers and the Committee on Research:

Report of Tellers

BALLOTS FOR OFFICERS

President—E. O. Eastwood	750
1st Vice-President—M. F. Blankin	733
2nd Vice-President—S. H. Downs	746
Treasurer—E. K. Campbell	743
Members of Council—L. G. Miller	748
A. J. Offner	745
A. E. Stacey, Jr	751
B. M. Woods	750

Scattering Votes for—*1st Vice-President:* W. E. Heibel, A. P. Kratz, A. J. Offner, J. H. Walker and C.-E. A. Winslow.

2nd Vice-President: M. F. Blankin, A. P. Kratz and H. Mathis.

Treasurer: J. F. S. Collins Jr., S. H. Downs, E. N. McDonnell, A. J. Offner, R. J. Tenkonohy and G. L. Tuve.

Members of Council: N. D. Adams, R. E. Backstrom, J. H. Carter, J. Davis, S. H. Downs, James Holt, H. J. Rose and J. H. Walker.

BALLOTS FOR COMMITTEE ON RESEARCH

C. F. Boester, St. Louis	751
J. A. Goff, Philadelphia	750
W. E. Heibel, New York	750
C. A. McKeeman, Cleveland	751
C.-E. A. Winslow, New Haven	750

Scattering Votes for—M. S. Wunderlich, G. L. Larson and W. M. Sawdon.

Total Ballots—801; Legal Ballots—751; Invalid Ballots—50.

BOARD OF TELLERS,
Joseph Wheeler, Jr., *Chairman.*
Harry H. Bond, Charles R. Hiers.

H. H. Erickson, Chairman of the Committee to Study Method of Selecting Society Officers and Council Members, presented his report which had been mailed to all members on January 14, 1942. Accompanying the report was an Amendment to Article B-VIII, Sec. 11 of the By-Laws offered by petition in accordance with Art. B-XVI, Sec. 1 of the By-Laws, copy of which had been mailed to all members under date of December 24, 1941. President Fleisher said that a discussion of the Committee's recommendations would be welcomed after which a vote would be taken upon the proposed Amendment to the By-Laws.

W. H. Driscoll, Syracuse, said that the Committee had studied many methods which it considered would improve the prevailing plan of Nominating Officers and Council Members. He said that every national scientific organization and all those in closely related fields selects its Nominating Committee by substantially the method proposed in the Amendment.

F. C. McIntosh, Pittsburgh, said that he approved the principle on which the Committee acted but considered that the amendment was defective and should have been submitted to the Committee on Constitution and By-Laws at least informally before it was circulated to members.

R. H. Carpenter, New York, commented on two points—the fact that over one-third of the membership did not have or want any Chapter affiliation and that the matter of expense was an important factor as the number of Chapters increased. He thought that the lack of representation by a large portion of the membership and the drain on the Society's treasury made a change in the Nominating Committee essential.

Albert Buenger, Cincinnati, said that he would like to bring out the fact that the present method of nomination resulted from a desire some years ago of Western chapters to have a stronger voice in Society affairs, and that his service on the Council convinced him of the absence of any sectionalism or chapter favoritism because Council members served for the benefit of the entire Society. He considered the present method cumbersome and thought that the special committee's suggestions should be adopted.

T. D. Stafford, Grand Rapids, said that he received no notification that the question of By-Law amendment would be considered at the meeting and he thought that the matter should be tabled.

President Fleisher said that the amendment and the report were mailed to individual members, not to Chapters, and asked for a show of hands of those who received a copy in advance of the meeting. (The majority of those present raised their hands.)

R. A. Miller, Pittsburgh, said that the present Amendment would in principle have his approval, but it would seem appropriate to refer the matter to the Committee on Constitution and By-Laws and have it properly presented by them after this discussion.

E. K. Campbell, Kansas City, said that the proposed amendment was similar to the method of nominating used some years ago by the Society and he considered that the present method of the nominating committee had produced highly satisfactory results.

John Howatt, Chicago, gave five specific criticisms of the Committee's report and read a communication prepared by the Board of Governors of the Illinois Chapter which expressed further opposition to the proposed amendment. He considered the change in method of nominating Officers and Council Members so important that it should not be acted upon until every member had a chance to study the matter fully.

L. T. Avery, Cleveland, thought that if engineering is deëmphasized in our Chapters, we will lose our appeal to the membership and if the Chapter is deëmphasized in the National Society, membership strength will be lost.

J. S. Kilner, Detroit, felt there were two distinct sides to the question concerning the importance of the national and chapter representation and he felt that some machinery might be set up where half of the Nominating Committee could be appointed by the Council and the other half by the Chapter Delegates.

C. E. Bentley, San Francisco, expressed opposition to the proposed amendment, especially because of the meeting attendance stipulation which was included among the qualifications for committee membership.

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C. Tasker, Toronto, said that members in Ontario were not satisfied with the present method of Nominating and he favored the change outlined in the proposed amendment.

J. W. Brinton, Boston, expressed the opinion of the Board of Governors of the Massachusetts Chapter that they unanimously favored the proposed By-Laws.

J. A. Bishop, Dallas, referred to a communication from the St. Louis Chapter and expressed the opinion that the present method should be retained.

Mr. Adams mentioned the fact that only 700 or 800 members voted for Officers and thought it would be interesting to find out how many of those who voted belong to Chapters. He also thought that the Nominating Committee should be named at the Annual Meeting each year by the members in attendance.

President Fleisher said that it would be necessary to adjourn at 12:15 P.M. for the Get-Together Luncheon, but the discussion could be continued at the afternoon session. He said that the By-Laws require a vote on the amendment at this meeting.

M. C. Beman, Buffalo, said that apparently consideration was being given to two methods of nomination and both were under fire, the present method because of the Committee's report and the Committee's suggestion is being criticized because it recommended substantially what we had once before. He made the suggestion that an entirely different method be substituted and have the Nominating Committee consist of the last five past presidents.

R. K. Becker, Evansville, said that as a non-chapter member he thought that there was room for improvement in the nominating procedure and the proposed plan would let many feel as though they were not total strangers in the Society.

Ferdinand Jehle, Indianapolis, suggested that, since an engineering society is probably no better than its chapters, he thought everyone should be assigned to a chapter and increase the dues to cover the costs involved.

C. F. Boester, St. Louis, said that the members of St. Louis Chapter voted unanimously to oppose the suggested change in the By-Law and he said that *ASRE* members were automatically assigned to chapter districts, so that there was no reason why this same method could not be followed. A luncheon recess was then taken.

Get-Together Luncheon

Following the morning session a get-together luncheon, which proved one of the most interesting, was held in the Clover Room for members, ladies and guests. Edgar A. Kirby, District Chief of the London Fire Force gave a very able and up-to-the-minute talk on Fire Raids on London, which he accompanied with a thrilling motion picture showing an actual raid on London, and just what should be done in an air raid. He gave a thorough explanation of how to handle incendiary bombs, and so that there would be no doubt in the minds of any in his audience, he illustrated incendiary bomb action with slides, which gave a clear picture of what to do and what not to do when handling incendiary bombs.

The discussion on By-Laws was continued at the afternoon session, and W. H. Driscoll, Syracuse, said that, no matter what results from this very animated discussion, we all have the utmost respect for the opinions of each other. He said the committee had given long and serious thought to the problem and it had deliberately offered a solution that they considered would cause the greatest controversy. He reminded members that this matter had been under consideration for three years, that there was dissatisfaction and discontent with the present arrangement, that the question had been much debated, but he thought a solution was in sight. He said that the whole matter had been handled under the procedure outlined by the Society's By-Laws and said the questions involved should be solved according to what the Society wants and it is up to the Society to make its decision.

Prof. C.-E. A. Winslow, New Haven, expressed the opinion that the present system is somewhat unwieldy and thought it would be unwise to press the matter to a conclusion, because it might create an unfortunate reaction. Professor Winslow offered the following motion, seconded by Professor Eastwood:

THAT: The proposed amendment to Article B-VIII, Section 11 of the By-Laws be tabled and that the special Committee's report and the transcript of the discussion from the floor be referred to the Committee on Constitution and By-Laws for consideration and for consultation by letter with the membership of the Society, and that the Committee recommend such appropriate action as it may deem desirable at a date sufficiently ahead of the 49th Annual Meeting to make action possible at that time.

President Fleisher called for a vote on the motion and it was carried.

An Evening of Magic

An Evening of Magic was held on Tuesday, January 27, when Paul Fleming Co. provided a two-hour program of legerdemain, mind reading, pseudo-spiritualism and illusions. It was an unique entertainment which included magic presented with dignity and gaiety with handsome stage settings and appropriate music. Paul Fleming is a magician who takes his art seriously and his work covers 25 years of study and platform experience. In *Who's Who* Paul Fleming Gemmill is Professor of Economics, University of Pennsylvania, and is the author of a number of books on economics, and is a member of many honor Societies.

Banquet

A feature of the annual banquet was the presentation of the F. Paul Anderson Award to Dr. F. E. Giesecke, professor emeritus at A. and M. College of Texas, College Station, Tex., and past-president of the Society. The presentation was made by Thornton Lewis, Army Ordnance Department, who created the F. Paul Anderson Fund, and who served as president of the Society in 1929. Dr. Giesecke received the medal in recognition of his notable contributions to the advancement of heating based on his research work in the fields of heat transfer and hot water heating and for his eminent services to

the Society as a member and officer. The F. Paul Anderson Medal Committee on Award for 1941 consisted of Prof. E. O. Eastwood, Seattle, Wash.; D. S. Boyden, Shirley, Mass.; W. T. Jones, Boston, Mass.; E. Holt Gurney, Toronto, Ont., Canada; and Prof. Perry West, Lexington, Ky.

W. L. Fleisher, New York, received the past-president's emblem from W. H. Driscoll, Syracuse, also a past president of the Society, and toastmaster A. J. Nesbitt, Philadelphia, introduced the newly elected officers: *President*—E. O. Eastwood; *First Vice-President*—Merrill F. Blankin, Philadelphia; *Second Vice-President*—S. H. Downs, Kalamazoo, Mich.; *Treasurer*—E. K. Campbell, Kansas City, Mo.

The speaker of the evening was Stanley High, nationally known commentator and writer. He served as an aviator in the first World War, and travelled extensively in Europe and is thoroughly acquainted with life in China, Japan and Korea. He was contributing editor to the magazine *Asia* and for four years edited the *Christian Herald*, after which he made a weekly radio broadcast. With his intimate connections in Washington and his knowledge of conditions in Europe and Asia his interpretations of domestic and foreign affairs have made him one of the most outstanding and popular commentators in America. He is a regular contributor to the *Saturday Evening Post* and *Readers' Digest*, and is the author of *A Waking World, China's Place in the Sun, Europe Turns the Corner*, and many others.

His subject was *What Are We Fighting for and Against?* In an inspiring address he stated:

"The present war is not primarily an economic war but rather a conflict of ideas. The fight is not to save the British Empire but to prevent it from becoming a Japanese or a German empire. Revenge should not be the driving force in the conduct of the war. Enough hate may give us a peace that will make the war not worth winning. That is why the common man recognizes that unsatisfactory as his lot may be now, in a democracy he has a chance to improve it. He sees that that chance goes when Hitler comes."

The seventh and final session was a joint session with the *National Warm Air Heating and Air Conditioning Association* and was opened by President Fleisher on Wednesday, January 28, at 2:00 P.M., who introduced C. A. Olsen, retiring president of the Association. After a few remarks by Mr. Olsen, George Boeddener, Managing Director of the Association, was introduced, as well as Allen W. Williams, the former Managing Director, who is also a Life Member of the Society.

R. R. Tucker, Smoke Commissioner for the City of St. Louis, gave an address on Smoke Control. He traced the history of the problems in the elimination of smoke from the City, which finally culminated in 1937 with the passage of a new ordinance which included a clause demanding that all fuel below 2 in. in size be washed until the ash content did not exceed 12 per cent. He stated this provision in itself did not attempt to reduce the smoke problem, but it was the beginning of the present program because it gave some measure of control over the type of fuel to be used. Subsequently, the ordinance was revised providing that all fuels should contain less than 23 per cent volatile matter on a dry basis, and if one desired to burn a high volatile coal, it was necessary to use some form of mechanical burning equipment. Mr. Tucker said that largely due to the cooperation of the citizens and many

varied interests of a community, the St. Louis smoke ordinance has been quite successful.

QUESTION: How do you handle the automatic firing of oil burners which are improperly adjusted?

MR. TUCKER: Our smoke regulations cover both oil burners and stokers.

MR. RHOTON: In the passage of this ordinance is it not true that people have actually saved money instead of putting them to additional expense?

MR. TUCKER: This is a controversial question. A survey conducted in St. Louis in 1929 indicated that the smoke problem cost the citizens about $19,000,000.00, which would mean that each individual was paying from $20.00 to $25.00 a year to live in the polluted atmosphere of smoke, which are admittedly hidden costs. In dealing with the general public it means little for me to say that we have reduced the soot deposit from 400 tons per square mile to 300, but if the soot is not on the window-sills or on the porches, then the home owner begins to believe that definite improvements have resulted.

The question of costs has been a very vital factor, primarily due to the extreme variation in the heat content of the fuel. Coals which are now being brought into St. Louis will vary from 13,000 to 15,000 Btu per pound, which means that there is 30 to 50 per cent more heat per pound of coal as compared to the fuel previously burned, which had heat contents as low as 9,000 Btu per pound.

L. D. PARSONS, JR., Washington, D. C.: I would like to inquire if you specify or control in any way the boiler or burner settings?

MR. TUCKER: Yes, this is done and in the case of power installations each one is considered a separate and distinct application.

R. A. SHERMAN, Columbus, Ohio: Basically Mr. Tucker is working on two principles for the elimination of smoke, one being the use of smokeless fuels and the other the use of equipment which will not produce smoke. Incidentally, the St. Louis smoke campaign has helped the stoker industry a great deal, but for the individuals in the lower income bracket who are forced to purchase fuels brought in from considerable distance, I am not sure whether these individuals feel that the increased cost has been justified.

In reality the effect of the St. Louis ordinance has been to awaken the bituminous coal industry to the fact that something must be done to utilize the available coals that lie within 10 miles of St. Louis, but which cannot be burned in the present automatic equipment. At the present time the coal industry has a program under way which is progressing to the point where some of this fuel can be burned smokelessly.

Installation of Officers

W. T. Jones, Boston, Mass., past president of the Society, conducted the installation of officers and those inducted into office were E. O. Eastwood, Seattle, Wash., *President;* M. F. Blankin, Philadelphia, Pa., *First Vice-President;* S. H. Downs, Kalamazoo, Mich., *Second Vice-President;* and E. K. Campbell, Kansas City, Mo., *Treasurer.* The newly elected Council members were L. G. Miller, East Lansing, Mich., A. J. Offner, New York, N. Y., A. E. Stacey, Jr., New York, N. Y., and B. M. Woods, Berkeley, Calif.

L. T. Avery, Chairman of the Resolutions Committee, offered the following resolutions:

Resolutions

WHEREAS, the 48th Annual Meeting of the ASHVE is to be concluded tomorrow without any further business session and

WHEREAS, the Technical Sessions, the Entertainment, Committee Meetings, and business transacted have been outstanding in interest and stimulation to those attending,

BE IT RESOLVED THAT an expression of thanks and appreciation be adopted and copies thereof be transmitted to each of the persons and agencies who have contributed to make this meeting so enjoyable;

To the National Society Officers and Committees and Past President Fleisher, who devoted so much of their time and energy to the work of the Society during the past year;

To the Officers and Members of Philadelphia Chapter and their ladies who have been so cordial in their welcome and so hospitable to everyone;

To Merrill F. Blankin, General Chairman and to Mrs. Blankin, Chairman of the Ladies Committee, who have been so considerate of our every wish;

To the Chairmen and Members of the several committees who have given so generously of their time to assure the successful carrying through of the Program;

To the authors and their associates who prepared and presented the fine technical papers, each of which is an outstanding contribution to the art and science of handling air;

To the Management and Staff of the Bellevue-Stratford Hotel;

To the Philadelphia Convention and Tourist Bureau for their cooperation;

To the Newspapers of Philadelphia for their fine coverage of this meeting and to the Trade Magazines who have cooperated so ably and generously;

To the Hon. Bernard Samuel, Mayor of Philadelphia and his official delegate, Mr. Charles Grakelow, for his inspiring speech of welcome to the City of Philadelphia;

To Edgar A. Kirby, District Chief of the London Fire Force, for his able speech and the interesting pictures given at our Monday luncheon;

To Paul Fleming and his associates for a marvelously mysterious Evening of Magic;

To Mr. Roth and his Associates who labored so arduously for the 7th International Heating and Ventilating Exposition which was postponed because of the war declaration;

To Past President Jones for his able installation of our new Officers; and BE IT FURTHER RESOLVED THAT in anticipation of an enjoyable banquet tonight we express our hearty appreciation to Stanley High and those who will make this an inspiring and a notable event; and

To all others who have contributed in so many ways to our enjoyment of the 48th Annual Meeting.

Respectfully submitted,
RESOLUTIONS COMMITTEE,
L. T. Avery, *Chairman*,
C. E. Bentley,
Brig.-Gen. W. A. Danielson.

At 4:45 P.M. on Wednesday, January 28, President Eastwood adjourned the 48th Annual Meeting of the Society.

Society Contributes to Victory Program

Believing that the national emergency requires an all-out effort from the engineering profession, the Society offered the full use of its facilities, including the specially trained personnel of its Research Laboratory, to the United States

Government without expense, and subscribed $40,000 of its funds to defense bonds. The resolution adopted was:

> THAT: *Whereas*, it is the unanimous belief of the Officers and Council of the American Society of Heating and Ventilating Engineers that an all-out effort in the national emergency should apply throughout the country, *Therefore, Be It Resolved*, THAT the Society offer without expense to the people of the United States through its Government, the full use of ASHVE Research facilities including the specially trained personnel of its Laboratory.

With over 100 of its members already in the armed forces of the United States, and many in the allied armies, there was definite indication of the important work which Society members are doing and have already done in safeguarding the efficiency and health of men in camps, industrial workers, and civilians.

48TH ANNUAL MEETING PROGRAM

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
THE BELLEVUE-STRATFORD, PHILADELPHIA, PA.

JANUARY 26-29, 1942

Sunday, January 25

11:00 A.M. Registration (Oak Room—18th Floor)
1:30 P.M. Council Meeting (Blue Room)
4:00 P.M. Final Meeting of 1941 Committee on Research (Blue Room)
7:45 P.M. Planetarium Trip

Monday, January 26

8:30 A.M. Registration (Oak Room—18th Floor)

9:30 A.M. BUSINESS SESSION—ASHVE Members only (Rose Room—18th Floor)
Welcome to Philadelphia by Charles H. Grakelow
Report of Council
Report of Treasurer
Report of Secretary
Report of Committee on Research, A. E. Stacey, Jr., *Chairman*
Report of Guide Publication Committee, C. M. Humphreys, *Chairman*
Report of Tellers of Election
Report of Committee to Study the Method of Selecting Society Officers and Council Members. H. H. Erickson, *Chairman*
Amendment to By-Laws

12:15 P.M. Get-Together Luncheon (for Members, guests and ladies) (Clover Room). *Subject:* Fire Raids on London—Edgar A. Kirby, District Chief, London Fire Force

2:00 P.M. TECHNICAL SESSION (Rose Room—18th Floor)
Panel Heating and Cooling Performance Studies, by B. F. Raber and F. W. Hutchinson
Radiation as a Factor in the Feeling of Warmth in Convection, Radiator and Panel Heated Rooms, by F. C. Houghten, Carl Gutberlet and E. C. Hach
Periodic Heat Flow in Building Walls Determined by Electrical Analogy Method, by Victor Paschkis
Heat Gain through Walls and Roofs as Affected by Solar Radiation, by F. C. Houghten, E. C. Hach, S. I. Taimuty and Carl Gutberlet

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8:00 P.M. TECHNICAL SESSION (Rose Room—18th Floor)
- Comfort with Summer Air Conditioning, by Technical Advisory Committee on Sensations of Comfort, Thomas Chester, *Chairman*
- Use of Cold Accumulators in the Air Conditioning Field, by R. W. Evans and C. J. Otterholm
- The Effect of Attic Fan Operation on the Cooling of a Structure, by W. A. Hinton and Albert F. Poor

Tuesday, January 27

9:30 A.M. TECHNICAL SESSION (Rose Room—18th Floor)
- Performance of a Hot-Water Heating System in the Research Home, by A. P. Kratz, M. K. Fahnestock, W. S. Harris and R. J. Martin
- Factors Influencing the Heat Output of Radiators, by A. C. Davis, W. M. Sawdon and David Dropkin
- A Comparative Study of the Friction-Heads in Screwed and Welded Elbows, by F. E. Giesecke and J. S. Hopper
- Design and Performance of Direct-Fired Unit Heater, by R. M. Rush and H. A. Pietsch

2:00 P.M. TECHNICAL SESSION (Planet Room)
- Corrosion Tests in a Water-Recirculating Air Conditioning System, by W. Z. Friend
- Entrainment and Jet-Pump Action of Air Streams, by G. L. Tuve, G. B. Priester and D. K. Wright, Jr.
- Determining Sound Attenuation in Air Conditioning Systems, by D. A. Wilbur and R. F. Simons
- The Performance of Stack Heads Equipped with Grilles, by D. W. Nelson, D. H. Lamb and G. E. Smedberg

6:30 P.M. Past Presidents' Dinner (Blue Room)

7:30 P.M. Meeting Chapter Delegates (Pink Room)

8:30 P.M. An Evening of Magic—Paul Fleming & Co.

Wednesday, January 28

9:30 A.M. TECHNICAL SESSION (Rose Room—18th Floor)
- Vapor Pressure of Ice from 32 to −280 F, by John A. Goff
- Physiological Fatigue from Exposure to Hot Environments, by M. B. Ferderber, M.D., F. C. Houghten and W. L. Fleisher
- Physiological Influence of Atmospheric Humidity, by Technical Advisory Committee on Physiological Reactions, C.-E. A. Winslow, *Chairman*
- Sanitary Ventilation, by W. F. Wells

12:15 P.M. Organization Meeting 1942 Committee on Research

2:00 P.M. JOINT SESSION with National Warm Air Heating and Air Conditioning Association (Rose Room—18th Floor)
- *Address:* Smoke Control, by R. R. Tucker, Smoke Commissioner, City of St. Louis
- Pressure Loss Caused by Elbows in 8-in. Round Ventilating Duct, by M. C. Stuart, C. F. Warner and W. C. Roberts
- Observed Performance of Some Experimental Chimneys, by R. S. Dill, P. R. Achenbach and J. T. Duck
- Heat Loss through Basement Walls and Floors, by F. C. Houghten, S. I. Taimuty, Carl Gutberlet and C. J. Brown
- Installation of Officers

7:00 P.M. ANNUAL BANQUET (Ballroom—Bellevue-Stratford)
- Toastmaster, A. J. Nesbitt
- Speaker, Dr. Stanley High, "What Are We Fighting For and Against?"
- Presentation of F. Paul Anderson Medal to Dr. F. E. Giesecke, by Thornton Lewis
- Presentation of Past President's Emblem to W. L. Fleisher, by W. H. Driscoll

Thursday, January 29

9:30 A.M. Organization Meeting of Council (Blue Room)
10:00 A.M. Inspection trips

COMMITTEE MEETINGS

Sunday, January 25

1:30 P.M. Council Meeting (Blue Room)
4:00 P.M. Final Meeting 1941 Committee on Research (Blue Room)
8:00 P.M. Technical Advisory Committee on Cooling Towers (Room 104)
8:00 P.M. Technical Advisory Committee on Sound Control (Room 105)

Monday, January 26

2:00 P.M. Technical Advisory Committee on Air Distribution and Air Friction (Room 104)
7:30 P.M. Committee on Code for Testing Air Cleaning Devices (Room 104)

Tuesday, January 27

9:30 A.M. Technical Advisory Committee on Air Conditioning Requirements of Glass (Room 105)
7:30 P.M. Technical Advisory Committee on Corrosion (Room 104)
7:30 P.M. Technical Advisory Committee on Radiation and Comfort (Room 105)
7:30 P.M. Technical Advisory Committee on Heat Requirements of Buildings (Room 106)
7:30 P.M. Chapter Delegates' Conference (Pink Room)
8:00 P.M. Technical Advisory Committee on Sensations of Comfort (Gold Room)

Wednesday, January 28

8:00 A.M. Technical Advisory Committee on Instruments (Room 104)
9:30 A.M. Technical Advisory Committee on Fuels (Room 105)
12:15 P.M. Organization Meeting 1942 Committee on Research (Blue Room)
2:30 P.M. Nominating Committee (Blue Room)
3:00 P.M. Technical Advisory Committee on Heat Transfer of Finned Tubes with Forced Air Circulation (Room 104)

Thursday, January 29

9:30 A.M. Organization Meeting of Council (Blue Room)

SPECIAL EVENTS FOR LADIES

Sunday

7:45 P.M. Leave Bellevue-Stratford for Planetarium

Monday

12:15 P.M. Get-Together Luncheon
3:30 P.M. Tea and Entertainment

Tuesday

9:30 A.M. Sightseeing trips to Historic Spots in Philadelphia and Luncheon at General Wayne Inn, Valley Forge
8:30 P.M. An Evening of Magic—Paul Fleming & Co.

Wednesday

10:45 A.M. Breakfast-Bridge (Red Room—Bellevue-Stratford)
Franklin Institute and Art Museum

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COMMITTEE ON ARRANGEMENTS

Philadelphia Chapter

M. F. Blankin, *General Chairman*

H. Berkley Hedges, *Vice-Chairman* — R. D. Touton, *Secretary* — C. B. Eastman, *Vice-Chairman*

Honorary Committee
(*Life Members*)

Homer Addams — J. D. Cassell
G. W. Barr — A. W. Luck
C. V. Haynes

Hospitality: H. H. Mather, *Chairman*
Hotels: Karl Rugart, *Chairman*
Publicity: J. H. Hucker, *Chairman*
Entertainment: W. P. Culbert, *Chairman*
Ladies: Mrs. M. F. Blankin, *Chairman*
Banquet: H. H. Erickson, *Chairman*
Transportation: L. A. Childs, *Chairman*
Inspections: E. H. Dafter, *Chairman*
Finance: A. C. Caldwell, *Chairman*
Sessions: R. F. Hunger, *Chairman*

No. 1192

PANEL HEATING AND COOLING PERFORMANCE STUDIES

By B. F. Raber * and F. W. Hutchinson,** Berkeley, Calif.

First progress report of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the University of California.

Introduction

THIS project covers a particular part of the radiant transfer problem which is as important to convective as to panel installations. In an earlier paper,[1] the authors developed a rational approach to panel analysis and discussed, from theory, the effect of the surface characteristic, emissivity, as one term in the basic radiation equation. The present paper reports on an experimental investigation of this emissivity factor and its practical significance in the design and operation of any type of heating system. Except where otherwise stated, the analysis, experimental results and general conclusions which follow apply with equal validity to warm air, convector, cast-iron radiator, or panel systems.

Heating systems, whether convective or radiant, have as their objective the maintenance of that relationship between effective air temperature, ET, and mean radiant temperature, MRT, which corresponds to optimum comfort. Each system has an effect, direct or otherwise, on both variables.

The possibility of realizing comfort in a room with air and surfaces at a uniform temperature of 70 F is well established. Likewise, the possibility of attaining comfort with air at 60 F and surfaces at 80 F has been reported.[2] European experience with panel cooling indicates that comfort can also be expected when air is 80 F and surfaces 60 F. In this connection convective heating and panel cooling may be considered similar because each establishes

* Professor of Mechanical Engineering, University of California. Member of ASHVE.
** Assistant Professor of Mechanical Engineering, University of California. Member of ASHVE.

[1] Panel Heating and Cooling Analysis, by B. F. Raber and F. W. Hutchinson, (ASHVE Transactions, Vol. 47, 1941).

[2] Physiological Reactions and Sensations of Pleasantness under Varying Atmospheric Conditions, by C.-E. A. Winslow, L. P. Herrington, and A. P. Gagge (ASHVE Transactions, 1938, Vol. 44, p. 179).

Presented at the 48th Annual Meeting of the American Society of Heating and Ventilating Engineers, Philadelphia, Pa., January, 1942.

comfort with air temperatures above 70 F and surface temperatures below 70 F; a like similarity holds between convective cooling and panel heating. The object of the present investigation was to determine the possibility of establishing comfort when both air and surface temperatures are below 70 F; experimental data are offered to substantiate the conclusion, valid from theory, that a room with both air and surfaces at 60 F can be comfortable, regardless

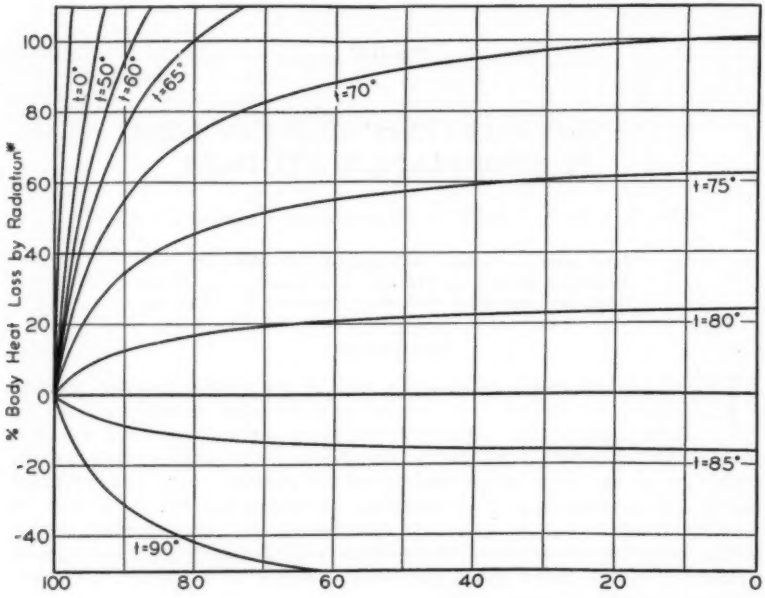

FIG. 1. TEMPERATURE-EMISSIVITY CHARACTERISTICS OF A 150 SQ FT ENCLOSURE

* The heat loss scale is expressed in per cent of that loss of body heat by radiation which occurs in a room with walls (e = 1.0) at 70 F and air at 70 F.

of the type of heating system employed, if certain surface and size requirements are satisfied.

PRINCIPLES

Within the limits of usual operating temperatures, conditions of comfort exist when ET and MRT bear a relationship to one another such that the body's sensible heat loss remains practically constant. At reduced air temperature the convective fraction of sensible loss increases, and therefore a corresponding decrease in the net loss by radiation becomes necessary. Heating systems of the panel type are usually designed to effect a decrease in the net radiant loss by increasing the emissive power of the surrounding surfaces.

The investigation covered by this paper deals with environmental systems in which a similar reduction of net radiant loss is accomplished by decreasing the effective absorptivity of enclosure surfaces and thereby reflecting back to the occupants some of the radiant energy which they emit.

As an example consider a room in which all surfaces are uniformly heated to 83 F (emissivity taken as 0.9) and the air temperature is at a value corresponding to comfort, approximately 57 F. The average surface temperature of the clothed occupants can be taken as 83 F [3] so there is no net radiant heat loss from the occupants. From an examination of this system one can readily conclude that, if the above conditions are comfortable, comfort will exist in any other environment for which the air temperature is 57 F and the net radiant loss is zero. Assume existence of a surfacing material having emissivity and transmissivity of zero (100 per cent reflectivity). Replace the 83 F enclosure with one having a 100 per cent reflective inside surface. Now, *regardless of the surface temperature,* there can be no net radiant transfer to or from the occupants and the flux in the enclosure must increase until the radiant energy being returned to the occupants after n reflections is equal to that which would be received if the enclosure were at a uniform surface temperature equal to that of the occupants, 83 F; conditions with respect to comfort are then the same as existed in the original enclosure with surfaces heated to 83 F.

While the comfort characteristics of the two enclosures described above are identical, the heating requirements are vastly different. In the first case a temperature difference of 26 F exists between the surrounding surfaces and the inside air; heat loss to the ventilation air would therefore be high. In the second case, the only heat required would be that needed to raise the ventilation air to 57 F and to supply transmission losses for the air-to-air temperature difference; the enclosure surface would come to an equilibrium temperature less than 57 F and the requisite heat input to the system could be achieved with any kind of heating system. With perfectly reflecting surfaces the size and shape of the room would not affect the condition of zero net radiant transfer.

NON-PERFECT REFLECTING SURFACES

A perfectly reflecting surface, as assumed above, does not exist. In extending the principle of *insulating* the occupants against radiant heat loss an examination is desirable regarding the extent to which existing materials can be employed to reduce net loss without need of increasing the emissive power of the surrounding surfaces.

Effect of Reflectivity

Fig. 1 shows the net radiant transfer between the occupant and the surroundings expressed as a percentage of that transfer rate which exists when walls are at 70 F and the room is large with respect to the occupant (effective emissivity of enclosure surface is then unity). The curves give per cent transfer as a function of enclosure reflectivity, for a given surface tem-

[3] THE ASHVE GUIDE, 1941, p. 762.

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perature, and are calculated for an occupant having 15.5 sq ft of effective radiating surface at 83 F.[4] To avoid the necessity of specifying shape factors, yet to obtain greater accuracy than is possible with the usual assumption that *the enclosed body is small with respect to the enclosure*, the curves are plotted from the exact analytical equation for concentric spheres or for concentric cylinders of great length.[5] The enclosure area is taken as 150 sq ft, equal

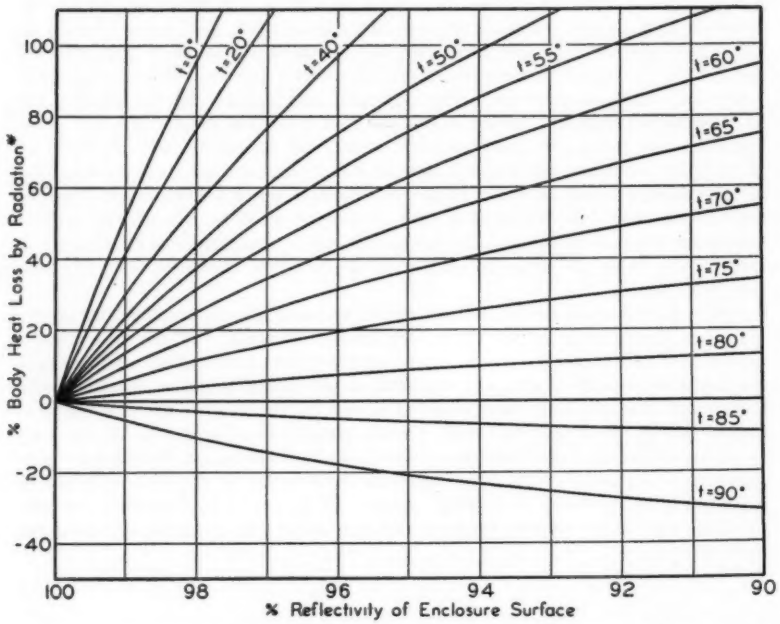

FIG. 2. TEMPERATURE-EMISSIVITY CHARACTERISTICS OF FIG. 1 INCREASED TO LARGER SCALE

to the area of the test chamber used in the experimental work. Reflection was considered as from perfectly diffusing (matte) surfaces.

Drawing a horizontal line through the 100 per cent transfer poiht of Fig. 1, the intersections with temperature curves give the respective reflectivities at which the net transfer from or to the occupants would be equal to that with a black body enclosure at 70 F. Interpreted in terms of comfort these reflectivities must exist for surfaces of the corresponding temperature if a condition of comfort is to be realized equivalent to that which would obtain if black body surfaces at 70 F existed. From Fig. 1 equivalent comfort would

[4] THE ASHVE GUIDE, 1941, p. 761.
[5] *Heat Transmission*, W. H. McAdams, Case 5, p. 54.

be realized in a room with air at 70 F and any of the following surface temperature-reflectivity combinations:

REFLECTIVITY PER CENT	SURFACE TEMPERATURE FAHRENHEIT
0	70
89	60
94	50
96	40
98	0
100	− 460 to infinity

The significant area of Fig. 1 is crowded into such a small section of the chart that this area has been redrawn (Fig. 2) to a much larger scale.

The complete curve for temperature as a function of reflectivity at constant net radiant transfer of 100 per cent (as defined for Fig. 1) is shown in Fig. 3. From the preceding tabulation and Fig. 3 it is evident that reflectivity becomes of practical importance only when its value is in excess of 90 per cent. Between 90 per cent and 100 per cent a small change in reflectivity results in a large variation in net transfer and therefore has an important effect on the necessary surface temperature.

Diffuse vs. Specular Reflection

The assumption of diffuse (matte-like) reflection corresponds closely to the condition of actual surfaces. If, however, *regular* or *specular* (mirror-like) reflection could be realized, the effectiveness of the surface as a form of insulation against radiant loss would be increased. Fig. 4 compares net energy transfer as a function of reflectivity for two surfaces, both at 70 F, one of which is 100 per cent diffusing and the other a perfect specular reflector. An inspection of the figure shows that the characteristics of both surfaces are identical for reflectivities of 0 per cent or 100 per cent and that as these values are approached the characteristics also draw together. In the reflectivity range from 40 per cent to 95 per cent the specular surface has a definite advantage. The reflectivity corresponding to maximum difference between the transfer rates occurs at that value for which the slope of the diffuse reflection curve is parallel to the curve for specular reflection; for the case shown in Fig. 4 this maximum difference is reached at 78 per cent. Inspection shows that the advantage of a specular reflecting surface is slight in the range of high emissivities, but is of great significance in the lower emissivity range which corresponds to polished metal surfaces.

Effect of Room Size

Perfectly reflecting surfaces cannot exchange radiation with the occupants regardless of room size. Perfectly absorbing materials do not permit reflections and hence establish an energy transfer that is likewise independent of room size. All other surfaces, however, absorb a fraction of the energy leaving the occupants and return the remainder. The fraction re-absorbed

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by the occupants depends not only on the emissivities of surrounding surfaces, but also on the number of re-reflections that take place between enclosure surfaces for each reflection which causes energy to strike the occupants. As the room size increases it is therefore evident that fewer of the infinite number of energy paths through the room will strike the occupants, hence the fraction of energy re-absorbed will decrease. Fig. 5, plotted from the same equation as Fig. 1, gives curves for energy transfer as a function of reflectivity for three enclosure areas all at the same temperature. The areas for which the curves apply are 900 sq ft (corresponding to a room 8 ft 6 in. x 14 ft x 15 ft); 150 sq ft (the area of the experimental enclosure); 16.5 sq ft (the

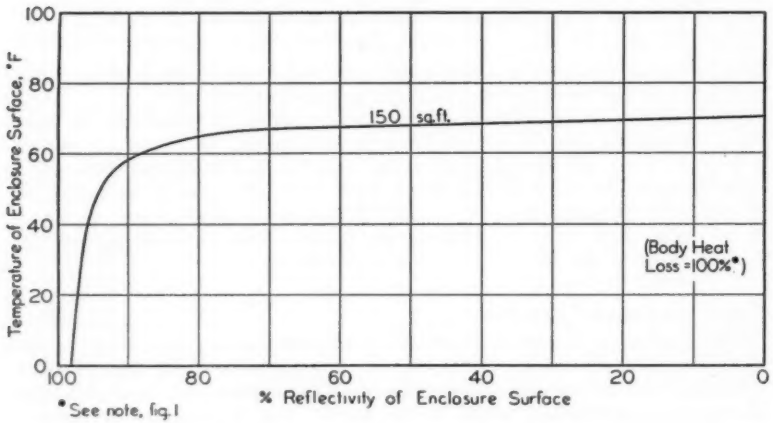

FIG. 3. TYPICAL TEMPERATURE-EMISSIVITY CURVES

approximate inside area of clothing for an individual of average size). Fig. 6 gives enclosure area as a function of reflectivity for a fixed surface temperature of 70 F and a constant heat transfer one half (or 75 per cent) as great as that from an occupant to a 70 F *black body* enclosure.

The effect of room size on the performance of non-perfect reflecting surfaces is evident from Figs. 5 and 6. As the room becomes smaller the equation expressing radiant transfer approaches the form used for exchange between infinite parallel planes; this latter equation is identical with that for the condition shown in Fig. 4 with specular reflection. Decrease in enclosure size therefore increases effectiveness by causing conditions in the space to more closely approach those which would exist if the surface were a specular reflector.

The discussion and curves of the preceding section bring out three characteristics of any enclosure designed to reduce, by reflection, the net radiant heat loss of the occupants. These are:

1. The effectiveness of reflectors is not a lineal function of reflectivity, but increases very rapidly as conditions of perfect reflection are approached.

2. Effectiveness increases as the surface approaches specular reflection.
3. Effectiveness increases rapidly as the size of the room decreases.

Evidently, therefore, an enclosure intended for experiment on reflection should be small and finished on the inside with a polished surface of low emissivity.

NUMERICAL EXAMPLE

Consider an enclosure 5 ft x 5 ft x 5 ft (150 sq ft of surface). If the reflectivity were 100 per cent, comfort of an occupant would be realized,

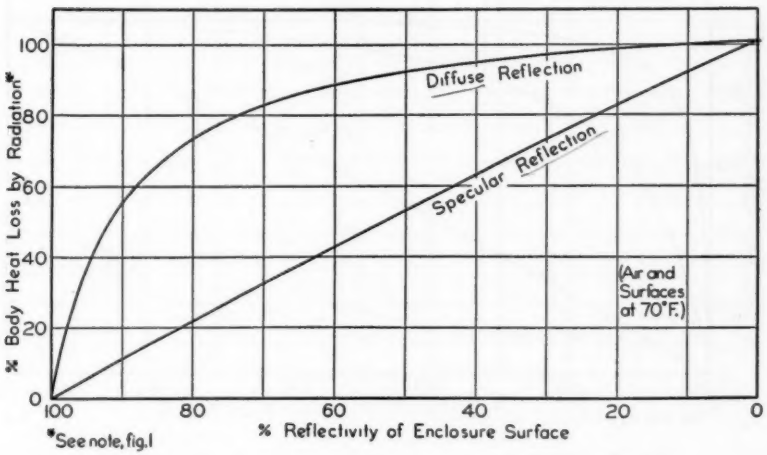

FIG. 4. SPECULAR VS. DIFFUSE REFLECTION IN 150 SQ FT ENCLOSURE

without energy input other than his own, when air and surfaces are at approximately 57 F. Since a real surface cannot be a perfect reflector the requisite comfort air temperature in an actual system must be sufficiently above 57 F to reduce the convection loss by an amount equal to the radiant exchange between surface and occupant. Referring to Fig. 2, the radiant loss to a 57 F surface having an emissivity of 0.025 (reflectivity of 97.5 per cent) is 41 per cent of that lost to a black body enclosure at 70 F. Black body loss at 70 F is 160 Btu per hour [6] so the radiant loss resulting from non-perfect reflection is 0.41 x 160 = 65 Btu per hour. Since the total sensible heat loss must be constant (for comfort), this 65 Btu per hour must be deducted from the convective loss. As a first approximation the convective loss may be considered directly proportional to the temperature drop from a clothed body (83 F) to

[6] THE ASHVE GUIDE, 1941, p. 65.

the ambient air; the necessary air temperature is then,

$$t = 83 - (300 - 65)\frac{83 - 57}{300} = 62.6 \text{ F.}$$

Thus in an enclosure 5 ft x 5 ft x 5 ft having perfectly diffusing surfaces of 97.5 per cent reflectivity, comfort conditions would be anticipated with air at 62.6 F and surfaces at 57 F.

If the surface of the above enclosure gave specular reflection, the radiant heat transfer would be 0.045 (interpolated from Fig. 4) x 160 = 7 Btu per hour and the requisite air temperature would then be,

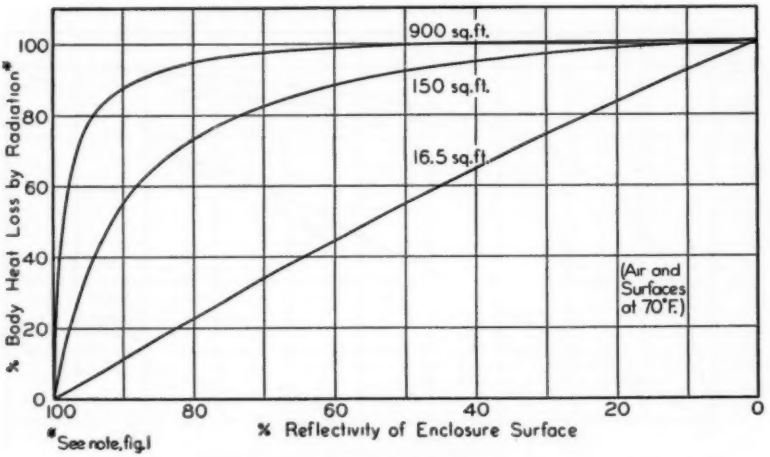

FIG. 5. EFFECT OF ROOM SIZE ON REFLECTION CHARACTERISTICS

$$t = 83 - (300 - 7)\frac{83 - 57}{300} = 57.7 \text{ F.}$$

Since an actual surface is a limited specular reflector its performance is somewhere between those just investigated and one would expect that the requisite air temperature would be within the above limits. Approximately, then, an actual polished surface enclosure 5 ft x 5 ft x 5 ft with emissivity near 0.025 would be expected to provide substantial comfort in air at 60 F when surfaces are 57 F.

EXPERIMENTAL SET-UP

Tests conducted in a small room accentuate the effect of reflection and hence increase the accuracy of the data. For this reason the reflection chamber designed for this project was made as small as practicable (5 ft x 5 ft x 5 ft). If actual comfort data from such a chamber agreed reasonably well with

theory the functional size effects noted in Fig. 6 could be readily used to extrapolate results to larger rooms.

Copper has a higher reflectivity than most commercially available materials so the chamber was constructed of 20 oz sheet copper with inside metal surface exposed and carefully polished. The exterior structural frame was made of ⅛ in. x 1 in. x 1 in. angle iron to which the copper sheets were attached by means of polished No. 4-36 brass machine screws. Access to the chamber was through one of the sides, hinged at the top. Ventilation was provided through cracks equivalent in area to inlet and exhaust openings each 6 in. x 6 in. The occupant *saw* 149.5 sq ft of reflecting surface and 0.5 sq ft of

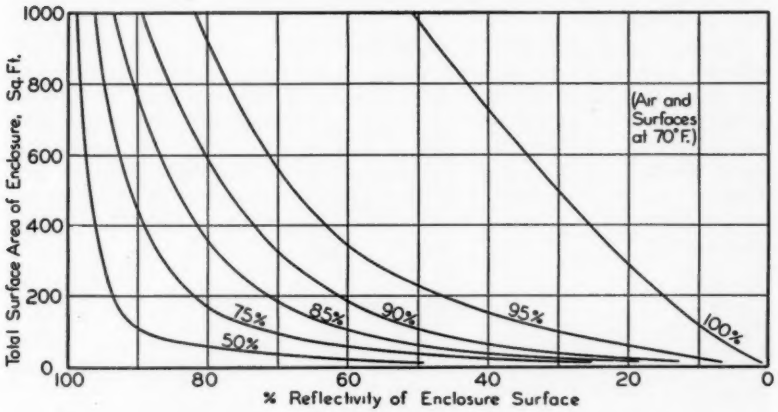

FIG. 6. TYPICAL SIZE VS. EFFECTIVENESS CURVES

crack. Taking the emissivity of the metal surface as 0.022 and the crack as a black body, the effective emissivity of the chamber was,

$$\frac{149.5 \times 0.022 + 0.5 \times 1.0}{150} = 0.025, \text{ or } 97.5 \text{ per cent. reflectivity.}$$

The actual test chamber therefore corresponded closely to the conditions of the numerical example already considered and, from theory, comfort would be expected with air at approximately 60 F and walls at 57 F.

The reflection chamber was placed on a raised platform inside one of the larger air conditioning research rooms, the exterior view of which is presented in Fig. 7. By means of equipment installed above the rooms the volume and temperature of the air passing through the larger room, and around the reflection chamber, could be controlled, and hence the surface of the reflection chamber or its contained air could be held at desired temperatures. The inside surface and the inside air temperatures are not independent variables, but for the chamber used in these experiments their relationship corresponded to the desired conditions. This can be shown, approximately, as on following page.

Sensible heat transfer from a comfortable occupant to enclosure approaches 300 Btu per hour. The equivalent inside surface film coefficient of heat transfer (for emissivity of 0.025) approaches 0.8 Btu per hour per square foot per degree Fahrenheit.[7] Then,

$$q = hA\ (t_a - t_s)$$
$$300 = 0.8 \times 150\ (t_a - t_s)$$
$$t_a - t_s = \frac{300}{150 \times 0.8} = 2.5\ \text{F}$$

Thus for an inside air temperature of 60 F the surface temperature would be 57.5 F which corresponds closely with the desired value of 57 F. Experiment showed an equilibrium overall temperature difference from inside to outside air of 3½ F.

Thermocouples were placed on inside and outside surfaces of the reflection chamber and in the inside and ambient air. After considerable experiment the most satisfactory method of measuring inside air temperature was found to be through use of a copper-constantan thermocouple inserted in a drilled and sealed hole, edgeways in a piece of 3/32 in. x ½ in. x ½ in. copper; the metal was selected to be free of scratches or surface imperfections of any kind and was then carefully cleansed and polished. Readings of this thermocouple checked those from a double-shielded, aspirating, calibrated, dry-bulb thermometer.

TEST CONDITIONS

The sixty subjects tested were largely volunteers from the junior and senior classes in mechanical engineering. With a few exceptions the age limit was from 19 to 23; 56 of the subjects were male.

Adoption of a standard of clothing did not seem feasible because of the relatively large number of subjects. In consequence, subjects entered the test chamber in the same clothing which they had worn throughout the day. Of those tested 20 wore neither sweater nor coat and had shirts open at the neck; 21 wore sweaters of varying weights. The remaining 19 were dressed in two- or three-piece suits, or outside jackets, or, in two cases, wore both a sweater and a coat. While variation in the manner of dress was strikingly great it none the less represented the selected conditions of different subjects for comfort in the same outside atmosphere.

The only heat source in the room, other than the occupant, was a 6-watt globe which provided enough light to permit reading, or writing, during the period of occupancy. Humidity was not controlled, but in many of the tests the subject was asked to operate a small shielded-bulb aspirating psychrometer and to record the wet- and dry-bulb temperatures at the 30-min interval; the average relative humidity was 65 per cent. The maximum effect on comfort dry-bulb temperature of an increase in relative humidity amounting to 15 per cent greater than the 50 per cent standard would be an increase of 1 F; in the present tests, however, increased humidity would have an opposite effect

[7] Radiation and Convection from Surfaces in Various Positions, by G. B. Wilkes and C. M. F. Peterson. (ASHVE TRANSACTIONS, 1938, Vol. 44, Fig. 2.)

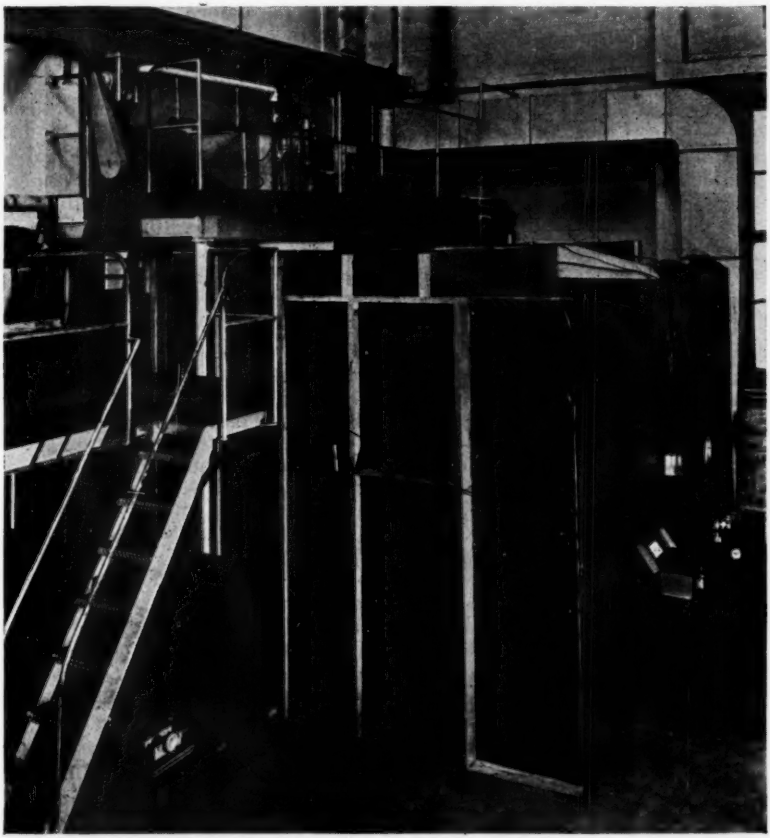

FIG. 7. EXTERIOR VIEW OF AIR CONDITIONING RESEARCH ROOMS

of undetermined magnitude due to absorption by water vapor of radiant energy, with a consequent reduction in the amount of radiation being returned to the occupant. These two contrary effects of humidity are both small, so it was considered satisfactory to assume that they compensated for one another and therefore could both be neglected.

All tests were run between the hours of 11 A.M. and 5 P.M. on days when the air temperature outside of the air conditioning research room was between 70 F and 80 F; for over 75 per cent of the tests this outside temperature was between 73 F and 77 F. The effect of entering a cooled enclosure from an atmosphere of higher temperature is one of slight shock, so the warm outside conditions were considered conservative, since their effect, if any, would be to cause the occupant to be somewhat cooler.

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The reflection chamber is exposed on all sides, top and bottom to the ambient air in the south air conditioning research room. To enter the chamber the subject was therefore obliged to pass through the ambient air which was approximately 3½ F lower (and at higher velocity) than the inside air in which each test was conducted. The average time of exposure to the ambient air was approximately 30 sec; in this interval the subject stepped into the enclosure and the side was returned to its closed position. As with the warm outside air, brief exposure to the colder ambient conditions should, if anything, have accentuated the shock and hence made the test results conservative.

A third acclimatizing factor existed in the necessity of entering an enclosure initially at a temperature 3½ F below that value assigned for the test. Since no heat was supplied to the enclosure its temperature at the start of each test was equal to that in the ambient space. However, the rise occurred rapidly following admission of the subject and within 4 min the air temperature was within 1 F of the desired value. Six minutes after occupancy started the air temperature had risen 3½ F and was at the value selected for test. The effect of this brief heating-up period would, again, be to slightly chill the subject and lead, therefore, to a conservative estimate on his part of the degree of comfort realized in the enclosure. Since the purpose of the tests was to investigate, experimentally, the effectiveness of reflection heating rather than to evaluate precise comfort reactions, it was considered satisfactory to neglect all such minor factors providing that, as in the three acclimatization effects just described, their omission would increase, rather than decrease, the difficulty of establishing comfort through reflection of body radiation.

TEST PROCEDURE

Instructions to Subjects

Before entering the reflection chamber each subject was asked to read the following instructions:

The purpose of the test which you are taking is to determine, statistically, the feeling of warmth experienced by an average individual when subjected to a particular type of environment. The physical data relative to conditions in the room during the interval in which you remain there are being recorded (from thermocouple readings) by the operator. After the test he will be glad to tell you the air temperature, wall surface temperature, etc., that existed during your period of occupancy.

The data which we ask that you record are subject to many variables and it may be that you will be unable to make a definite statement as to whether you are cool, warm, cold, hot, or ideally comfortable. In this event we ask that you record the best possible estimate.

Conditions in the room will be varied for different subjects so it is important that you do not enter with the preconceived idea that you *should* be comfortable, or cold, or hot. In some cases conditions will be changed during a test so that you may be uncomfortably warm at the start and cold before the test is over. Therefore you should record your general feeling of warmth at intervals of 5 min.

To establish a reasonable uniformity of reports you are asked to report your feeling of warmth, by number, from the following schedule:

1. Uncomfortably hot.
2. Uncomfortably warm.
3. Comfortably warm.
4. Ideally comfortable.
5. Comfortably cool.
6. Uncomfortably cool.
7. Uncomfortably cold.

The preceding schedule refers to your *over-all* degree of comfort. In addition you may notice local conditions as drafts, heating or cooling of exposed skin, feet, back, etc.; after recording the number which defines your *over-all* feeling of warmth write any comments or observations on conditions which you consider unusual or significant.

After reading the instructions each subject was provided with a pencil, watch, and a standard form for recording his observations. The first reading was requested as soon as the reflection chamber door had been closed and subsequent readings at five minute intervals thereafter.

Position of Subject

The only furniture in the reflection enclosure was a small stool and two ¾ in. foot pads, of insulating material, arranged to be moved to that position on the floor for which the occupant was most rested. In entering the enclosure the subject was warned not to step on or touch with hands the reflecting surface. The subject was seated on the stool with feet resting in a comfortable position of his own selection on the foot pads.

Throughout the test the occupant was obliged to remain in a sitting position and was not free to alter his position other than through slight movement of the trunk or crossing of the legs.

Record of Data

The average test was for a period of approximately 50 min. Consideration was given to a longer test period, but other demands upon the time of the subjects made it improbable that a sufficient number of volunteers could be obtained for any interval longer than that of a regular University class period.

The possibility of serious acclimatization error due to the short test period was investigated by testing for a 2½-hour period three subjects; in every case the results from the longer tests corresponded exactly with data from 50-min tests of the same subjects under the same temperature conditions. In many of the later tests the subject's sensitivity to changing conditions was investigated by varying the temperature during the last 10-min interval. To permit expressing data from all tests on the same basis, and for a period during which conditions remained unchanged, the subject's reactions for the first and last 10 min intervals were not used for analysis, but four readings from consecutive 10-min intervals in the center of each test were selected.

As soon as the occupant was in the reflection enclosure the operator started recording temperatures and continued to make a complete record of all temperatures throughout the system at 5-min intervals during the 50-min test period. All temperatures were obtained from No. 36 copper-constantan thermocouples connected through a control board to a precision potentiometer which could be read to 0.001 millivolts. The potentiometer was checked against a standard cell at intervals during each test. Additional checks on temperature readings outside of the enclosure were available from a series of resistance thermometers and triple, 24-hour, recording thermometers.

Control of the ambient temperature was manual by adjustment of a throttling valve in the dichlorodifluoromethane suction line just ahead of the compressor; accurate control of the air temperature to within ± 0.5 F of the desired setting was realized without difficulty. To maintain uniformity of external film coefficients on the enclosure and hence to keep the air-to-air temperature drop from

varying, the volume of air circulated through the ambient space was held constant for all tests.

The psychological effect of divorcing the occupants from any direct knowledge of the existing air temperature was investigated, but no correlation was obtained between the feeling of warmth and the occupant's knowledge of air temperature. The conclusion was reached that the psychological factor, if existent, is too small to be of practical significance. All data from all tests were therefore used, without reference to the subject's knowledge or ignorance of the unusually low temperature in the enclosure.

EXPERIMENTAL RESULTS

Of the 60 subjects used in this investigation, fifty were tested at enclosure air temperatures between 55 and 65 F. Of the tests, 25 were conducted in air

TABLE 1—RESULTS OF TESTS BASED ON 4 CONSECUTIVE VOTES FROM REPORT OF EACH DIFFERENT SUBJECT[a]

SCHEDULE NO.	SENSATION	VOTES OUT OF 200	PER CENT OF TOTAL
1	Uncomfortably hot	0	0.0
2	Uncomfortably warm	1	0.5
3	Comfortably warm	4	2.00
4	Ideally comfortable	94	47.0
5	Comfortably cool	83	41.5
6	Uncomfortably cool	18	9.0
7	Uncomfortably cold	0	0.0
		200	100.0

[a] All votes for reactions Nos. 3, 4, or 5 represent comfort conditions, and 90.5 per cent of all votes were for comfort.

at ±1 F from 60 F and the remaining 25 were distributed over the ±5 F range in such a way that the weighted average for the 50 tests remained at 60 F. Use of a weighted average assumes a linear variation of the feeling of warmth with temperature; within the range considered and noting that 50 per cent of all tests were very close to the average temperature it is probable that this assumption is accurate within the limits of experimental error. The temperature of the enclosure inside surface was, in all tests, 3-4 F below the air temperature.

Results of the 50 tests based on 4 consecutive votes from the report of each different subject are shown in Table 1.

The large block (41.5 per cent) of *comfortably cool* votes indicates that respiration of air at the lower temperature causes the average person to be conscious of a pleasant coolness even though the exterior body heat balance is perfectly normal. Many of the subjects recorded the impression that the air in the enclosure seemed to them unusually *fresh* and invigorating; some of the subjects read or studied between readings and commented, on leaving, that the enclosure atmosphere did not induce the feeling of drowsiness which

they frequently associated with such activity. In contrast to the 22 subjects who commented favorably on the effect of breathing enclosure air, two men reported that the air seemed *stuffy*.

Analysis of Reactions by Subjects

The average reaction of each subject was determined from his four votes; 86 *per cent of all subjects had an average which represented conditions of comfort.* The group average for 50 subjects, one test each, was 4.57; that is, half way between the reaction scheduled as ideally comfortable and that scheduled as comfortably cool. Investigating the distribution of votes cast by each individual, the variation of reaction during the test period is found to be as follows:

Of 50 subjects,

76 per cent were comfortable throughout the entire test period.
12 per cent were comfortable for 75 per cent of the time and uncomfortably cool or warm for the remaining quarter of the test.
12 per cent were comfortable for 50 per cent of the time and uncomfortably cool for the remaining 50 per cent.
0 per cent were uncomfortably cold or hot at all times.

100 per cent

Discussion of Results

A comment repeatedly occurring on subjects' data sheets is that an immediate sensation of warmth was experienced when the door of the reflection chamber closed. Many subjects noted a feeling of warmth on all exposed skin surfaces while some stated that this effect was limited to the face. A few tests were made with subjects having rolled sleeves; they reported that the exposed forearm remained comfortably warm for the duration of the test.

The most common report of local cooling was with respect to the ankles and feet of the male subjects. This was attributed to a draft near the floor of the enclosure, but this could not be detected with instruments. Tests of 4 female subjects disclosed an opposite reaction and suggested the possibility that the trouser leg may largely shield the ankle from radiation while exposing it to the cool air. Additional tests to investigate this hypothesis are in progress.

Data from Other Tests

Investigations with temperature conditions above or below those for comfort were not included in the objectives of the present work, which was concerned with comfort area. However, a few subjects were tested with enclosure air temperatures in excess of 65 F. The reactions they recorded, at temperatures above 66 F, included decided feelings of *stuffiness*, and, at 68 F, activity of the sweat glands became noticeable in two cases.

Six subjects were tested in air at 57 F with all surfaces at 52 F and with a supplementary heat source of 100 w; all these subjects were comfortable at all times, but the number tested is too small to allow consideration of these results as representative. Further, these six tests were performed under conditions which did not permit determination of the per cent radiant transfer from the 100-w source. Since the convective fraction from such a source would not in any way influence the occupant's feeling of warmth (air tem-

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perature was controlled by the operator and the convective fraction represents added cooling load), an analytical heat balance for this case is not possible.

CONCLUSIONS

Inside surfaces of low emissivity (high reflectivity) can be effectively used to reduce the heating requirements of an occupied room. This reduction is independent of the type of heating system.

Effectiveness of reflective walls varies inversely with room size and decreases rapidly with a reduction in the reflectivity of the surface. The equivalent reflectivity of a room may be low even though a large part of the inside surface is covered with highly reflective material; small *black body* areas, as open windows, door or grilles, severely reduce the equivalent reflectivity of the enclosure.

If polished surface insulation, as metal foil, is to be used in a given structure, and if architectural or other considerations will permit, an additional advantage will accrue from using the material on the inside surface rather than in the walls. When used in the interior of walls, reflecting surfaces reduce transmission losses; when used as an inside surface finish, the same materials not only reduce transmission losses, but act also to prevent net heat loss from the body and thereby permit realizing comfort with a lower air temperature and a reduced heating load.

Experimental data from 50 tests with 50 subjects established the possibility of attaining comfort in a reflecting enclosure with air at 60 F and surfaces at 57 F and with no heating of any kind other than that due to the occupants.

DISCUSSION

C.-E. A. WINSLOW, New Haven, Conn. (WRITTEN): I have listened to the paper by Professor Raber and Mr. Hutchinson with great interest and have had the privilege of visiting the laboratory at Berkeley and seeing the experiments in operation.

The authors' theoretical analysis of the facts which determine the effectiveness of re-radiation surfaces is most valuable and should be basic in our thinking about this complex and fascinating problem; and their actual experiments in the reflection chamber demonstrate the practical significance of the procedure. I have been working in this field now for a good many years but nevertheless felt a real thrill on sitting down in Professor Raber's copper booth in cold air and feeling waves of heat come to my face from the cold copper surfaces as a result of re-radiation of my own body heat. The sensation is in kind if not in degree similar to that experienced on opening the door of a furnace, and it gives one a vivid sensation of the efficacy of the human body as a heat source.

I should like to suggest a word of caution in regard to a possible misinterpretation of the results presented. It is quite true, as the authors say, that in a booth of high conductivity and with relatively free circulation of cold outside air within the enclosure, the human body can only keep itself warm in a booth of small size. In an actual building, with any local heated surface, instead of the human body, as the source of radiation, the same general principle applies. With such conditions, however, with heat conductivity reduced to a relatively low level and no considerable inflow of cold outside air, a state of equilibrium is ultimately reached between heat input and heat output. The larger the room, the longer it takes to reach such a state. After equilibrium is attained, however, air and wall temperatures throughout the space ap-

proach each other closely. When this occurred the saving effected by re-radiation will still, as the authors have theoretically demonstrated, bear a relation to the size of the room; but the quantitative factors will be such that economy due to the use of surfaces of high radiation-insulating value remains substantial and important.

T. NAPIER ADLAM, Bethlehem, Pa. (WRITTEN): Before making any comment on this very interesting paper I should like to thank the authors for what I consider a very valuable contribution to the science of human comfort.

The results of their tests and the very careful expounding of the principles on which the tests were made make very interesting reading, and it helps to substantiate something we have experienced for a very long time; *viz,* that unheated surfaces with high reflectivity give much better results than surfaces with high absorptivity. The heat loss is made less and the comfort much greater. So far, however, the difficulty has been to obtain suitable material with surfaces which, while having a high reflectivity, will also harmonize with the interior decoration. In reviewing these tests and the conclusion arrived at, one must, I think, bear in mind that under normal conditions it will not be possible for occupants of a room to be exposed on all sides to these high reflectivity surfaces; I feel sure that a larger room would give a much different result. One cannot assume that people will occupy a room without furniture; and articles of furniture will naturally absorb energy leaving the body either by direct rays from the body or by the rays reflected from the surrounding surfaces of high reflectivity. The articles of furniture having absorbed some of the energy will then give off some heat to the surrounding air as convected currents. Some of the energy received by the articles of furniture will, of course, be reemitted as longer wave-lengths and, therefore, one must assume that in a general way persons occupying a room where these specially prepared surfaces are used will be exposed both to surfaces having high reflectivity and to surfaces emitting energy at a somewhat lower temperature. It may be assumed that gradually the articles of furniture will take up the same temperature as the air, but unless the furniture has the same high reflectivity as the special walls and ceiling, these will absorb some of the energy from the human body which will, of course, not be returned to the body. While the temperature of 83 F is assumed to be correct as the average surface temperature of a person under predetermined conditions, it would be interesting to know if this was checked on the occupants during the test.

If a person is placed in a room with an air temperature of 70 F, the surface of the clothes will be one temperature, but in a room with an air temperature of 57 F, the surface temperature of the clothes will be something different. In other words, the temperature gradient from the skin surface of the body through the clothes will be much steeper. It is I know possible to balance the heat loss due to cooler air by radiant panels, but if we attempt to rely on high reflectivity surfaces only, I think we should consider this point very carefully and check same during any such test. As regards the feelings of those taking part in the test, it would be interesting to learn if those wearing outdoor clothes or warm sweaters felt more comfortable than those wearing shirts opened at the neck. Some persons will freely expose themselves to outside conditions and become accustomed to cool temperatures without any feeling of discomfort, whereas others, who protect themselves very carefully when outside, might find a cool temperature very uncomfortable unless it is balanced by some heated panels. If the surfaces used had a full 100 per cent reflectivity, one would not expect much feeling of discomfort from younger people, except from the fact of sitting still for 50 min., but with older people I am afraid there will be a larger percentage whose feelings would be on the chilly side. If a series of similar tests could be taken with elderly people it would be very interesting to compare the results, although I think to ask an occupant to sit on a stool in one position for 50 min without moving is somewhat abnormal, unless their attention can be diverted by something very interesting. When we are dealing with these problems we have to realize that elderly people require comfort and a little more protection from heat loss than the younger

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ones, especially when they reach the stage where the fire within burns more slowly. I would like to suggest such a test to the authors.

It is interesting to note that a common report was of a cooling effect at the ankles and feet of the male subjects. This may be partly due to the fact that the leg would be shielded from rays by the trousers and that a created convected current of cool air may chill the ankles and feet, but, on the other hand, I think that since females are accustomed to unprotected ankles except for thin stockings, regardless of the weather conditions, they would not feel the effect of the cooler air as would the male. When a male is walking or standing in a cool atmosphere, his trousers reach to his shoe and protect his ankle; but when in a sitting position, as was the case when the test was made, his trousers are lifted leaving his ankle exposed to the cold air. One would, naturally, expect males to feel some discomfort because of this. It is, of course, quite possible that the air near the floor was somewhat cooler than at a little higher level, and the females with high heels would have their ankles raised higher than the male. If the floor had been warmed as with radiant heat, I do not think any of these discomforts would have been felt.

The subject and treatment of the paper is extremely interesting because there is a lot to be done in the way of choosing the most suitable surfaces to give greater comfort and save fuel.

H. B. NOTTAGE, Hartford, Conn.: I believe the authors should be commended for the method of attack which was used in the analysis. The problem was assumed and next a certain physical system was established and then a hot body placed in it and finally by a simple heat transfer analysis the effects of emissivity were predicted. Later a subject was selected for analysis and measurement, which made it possible for the investigation to verify the preliminary predictions. This sort of procedure is well worth while and I think the results, while they are not final, are a step in the right direction.

L. T. AVERY, Cleveland, Ohio: These tests seem to indicate that in order to make a person feel comfortable with heated air, a wall surface is used having a high reflectivity to heat. I would like to inquire how this condition can be reversed when it is desired to use cool air.

MR. HUTCHINSON: This question can be answered by assuming that a room has perfectly reflecting walls. In that event, the occupant during the heating season could not lose energy by radiation to such walls. During the cooling season, if this room was equipped with a 1 sq ft panel having an emissivity of 1.0, then the total cooling load in such a room would consist only of that heat necessary to maintain body comfort for the occupant. If in such a theoretical room perfectly reflecting surfaces were possible one could maintain optimum summer comfort with a total cooling load of approximately 50 to 60 Btu per occupant; if the number of occupants were to double, the cooling load would also double. In other words, the load is a function of occupancy rather than of transmission losses. Hence, in a theoretical case, the system is equally applicable to summer or winter conditions.

F. C. HOUGHTEN, Pittsburgh, Pa.: This is a very interesting paper on a timely subject, and it involves two questions in which the ASHVE Research Laboratory has been interested. One of these deals with the effects produced in an enclosure with panel heating and the other is the effect of this radiant energy on a person's feeling of warmth. This paper gives the effect of a 1 deg increase in mean radiant temperature, as equivalent to a 1 deg decrease in dry-bulb temperature, which is not entirely in agreement with the results obtained at our Laboratory.

MR. HUTCHINSON: When this investigation was started we were not interested in the comfort curve and therefore we utilized such data as were available. It was assumed that comfort could be realized at approximately 57 F air temperature with wall temperature of 85 F. As mentioned in the paper we found that 90.5 per cent of the votes were comfortable. If we had investigated different conditions and our assumptions had been incorrect, then the number of comfort votes would have varied

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considerably. However, the data obtained seem to justify our theoretical conclusions. This study was not an attempt to investigate the comfort curve, but simply to work with assumed data and determine whether the principle of reflective surfaces would operate.

W. L. FLEISHER, New York, N. Y.: Will you please amplify your statement that the so-called panel heating system is no more essential to obtain the results reported in your paper than other methods of heating? I do not believe that you brought out that point particularly clearly, as most of us have considered panel heating as something buried in the ceiling or wall or floor.

MR. HUTCHINSON: The test rooms used for this investigation did not have any panels installed, but the inside surfaces were made of reflective material. The only heat supplied was by the test subject placed in the enclosure who artificially maintained the ambient temperature due to heat loss from the body. The copper shield, or test enclosure, could have been taken outdoors where the air temperature was approximately 40 F and the subject placed in it and equilibrium temperature allowed to be realized. In effect, we had no heating system of any kind except that of the human body. The important point to remember is that in such a room if the air temperature is brought up to 57 F by means of a radiator or convector equally good results could be obtained.

W. R. RHOTON, Cleveland, Ohio: I am interested in this particular subject because some of these principles were used in designing the notorious bed which you have heard me previously talk about. In this application mirrors were placed on the inside and while sleeping it was found that under such conditions it was not necessary to supply heat, but actually supply cooling to produce comfortable conditions. I think there are sufficient data available at the present time for most engineers to design a radiant heating system not only for comfort but for various types of industrial applications.

C. F. BOESTER, Lafayette, Ind.: At the present time we are doing quite a lot of work on this subject at the University of Purdue in cooperation with Dr. C. A. Mills of Cincinnati. We are now equipping a six-room residence to work out some of the theories that are outlined in this paper. Several techniques have been developed for obtaining surfaces of high reflectivity, including some special plastics that can encase evaporators so that fractional horsepower refrigeration equipment can place a cycle in reverse during the summer. Different devices for producing heat radiation in the winter have been studied, including methods of diffusion so that regardless of the location of the individual there will be a balance of heat loss produced. At the present time we feel that the air inside the room should be about the same temperature as the outside air. For instance, if it is 0 outside today it might logically be desirable to have 0 temperature inside the room to offset any body loss by convection, etc. We would add radiant energy with primary control being obtained with reflective surfaces.

P. NICHOLLS, Pittsburgh, Pa.: This paper strikes me as a study concerned with saving heat to the body by radiation instead of by clothing, and consequently, one might wear lighter clothing in a house which is equipped with radiant surfaces.

E. M. MITTENDORFF, Chicago, Ill.: Is there any reason to assume from these tests that if the same surfaces were used in the floor and the ceiling the same results would be obtained?

MR. HUTCHINSON: The important factor is not where the surface is, but the resulting equivalent surface emissivity of the entire structure. We were working with an equivalent inside emissivity of 0.025. If a ceiling reflective surface is added to a room and the wall is of non-reflective surface then the equivalent emissivity would be quite high.

J. B. FULLMAN, Pittsburgh, Pa.: This paper deals in an interesting manner with a problem recently encountered by a designer with whom I discussed a floor type heating system in a building having sheet metal interior surfaces. We wondered what the

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effect of the polished metal surfaces on the air temperature required for comfort would be.

The building had a floor area of about 600 sq ft and when new, the metal surfaces would have an emissivity of about 14 per cent. The weighted average emissivity in the room, taking the floor area at an emissivity of 90 per cent, would be in the neighborhood of 35 per cent. The reflectivity would be about 65 per cent. Fig. 5 of this paper indicates that the reduction in heat loss from the body by radiation resulting from the use of these reflective surfaces would be something less than 10 per cent. Since normal radiation losses are approximately 200 Btu per hour, this would mean about a 20 Btu reduction in heat loss by radiation from the body. The normal convection loss is about 110 Btu per hour, and to compensate for the change in radiation loss, this would have to increase to 130 Btu per hour.

Normally, the temperature gradient from the surface of a person's body to the air is in the neighborhood of 15 deg. To increase the convection loss from 110 to 130 Btu would require about an 18 per cent increase in this body to air gradient or a change from a 15 deg gradient to an 18 deg gradient. Thus, if comfort were obtained in a conventional room at 70 deg, by lining the walls and ceiling with sheet metal, comfort conditions could be obtained at 67 deg air temperature.

This computation leads me to believe that actual air temperature reduction, as a result of the use of reflective interior surfaces, is going to be a minor factor in heating system design in spite of the rather striking results obtainable in special cases.

MR. HUTCHINSON: A partial answer to this question could be obtained by calculation and I will be glad to help Mr. Fullman obtain the answer desired.

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RADIATION AS A FACTOR IN THE FEELING OF WARMTH IN CONVECTION, RADIATOR AND PANEL HEATED ROOMS

By F. C. HOUGHTEN,* CARL GUTBERLET ** AND E. C. HACH,† PITTSBURGH, PA.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in its Research Laboratory at the Pittsburgh Experiment Station of the U. S. Bureau of Mines.

THE part which radiation plays in a person's feeling of warmth is receiving increased attention in modern heating and air conditioning applications. A research program outlined by the ASHVE Technical Advisory Committee on Radiation and Comfort [1] resulted in the building of two rooms at the ASHVE Research Laboratory in which studies were made during the past two heating seasons.

A report [2] on the results of the first season's study, when one room was heated by radiators and the other by a warm air circulating system, was presented to the 1941 Annual Meeting of the Society, and gave the relation between the mean radiant temperature and the other environmental conditions for the two rooms. As was to be expected, the mean radiant temperature was found to be higher in the radiator-heated room; however, in no test did it exceed the dry-bulb temperature of the air at the 30-in. level. The paper presented evidence on the relative effects of radiation and air temperature on a person's feeling of warmth.

In order to study this relationship when radiation was a larger factor in the heating effect of the room, and more particularly, when the mean radiant temperature exceeded the dry-bulb temperature of the air, a continued program, including the installation of ceiling heating panels in one of the two rooms and the same convection room arrangement in the other, was developed

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[1] J. C. Fitts, *Chairman;* A. H. Barker, L. M. K. Boelter, R. E. Daly, J. B. Fullman, E. R. Gurney, L. N. Hunter, A. P. Kratz, C. S. Leopold, D. W. Nelson, W. J. Olvany, G. W. Penney, W. R. Rhoton, C. J. Stermer, C.-E. A. Winslow.

[2] ASHVE RESEARCH Report No. 1172—Radiation as a Factor in the Sensation of Warmth, by F. C. Houghten, S. B. Gunst, and J. Suciu, Jr. (ASHVE TRANSACTIONS, Vol. 47, 1941.)

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and executed during the 1940-41 heating season. The results of this continued study are the basis of this report.

A view of the two test rooms on the roof of the warehouse building of the Pittsburgh Experiment Station of the U. S. Bureau of Mines is shown in Fig. 1, while Fig. 2 is a floor plan of the two rooms, one equipped with an air circulation or convection heating system and the other with a panel heating system. The construction and arrangement of the rooms as reported earlier was not altered, except that heating panels were installed in one room and the ceiling of each room was insulated.

The panel was constructed by placing four serpentine coils, each containing 70 ft of ½-in. standard wrought-iron pipe, against the structural insulation board ceiling of the room as earlier reported. Expanded metal lath was attached to the lower side of the coils and the entire assembly was anchored securely to the joist above with No. 8 wood screws. Cement plaster was then

FIG. 1. VIEW OF RADIATION AND COMFORT TEST BUILDING

applied, special precaution being taken to see that it completely filled the voids between the pipe and the ceiling. Some dimensional details and arrangements of the two 6 ft 4 in. x 10 ft panels on the 12 ft x 15 ft ceiling are shown in Figs. 2 and 3.

The panels were heated by a hot water system shown in Fig. 3 consisting of an insulated heater containing four bayonet heaters screwed into 2-in. pipe sections of the circuit, a hot water circulating pump, and 1-in. supply and return mains to and from the four panel coils. In order to continue the practice used in the earlier study of having as nearly as possible all of the electrical heat application within the rooms themselves, the electrical water heating unit, occupying about 1¼ sq ft of floor area, was located in the corner of the room. Supply valves gave individual control of the hot water to each of the four coils in the two ceiling panels. Mercury thermometers located in pipe tees so that their bulbs were in the water stream gave the water temperature leaving and returning to the heater. The electrical heat input to the four bayonet heaters was so arranged that part was under manual and part under thermostatic control. The part under thermostatic control was actuated by placing the sensitive element in the hot water supply line to the coils so as to maintain a constant water supply temperature. The temperature called for by the thermostat was raised or lowered by hand so as to have the desired effect on the room air. Either the manually or thermostatically controlled portion of the electric heater could be adjusted by six steps through

series or parallel arrangements; this gave a convenient and wide range of control of the heat input. Arrangements were made for observing temperature at the points indicated in Fig. 4. Unless otherwise stated the dry-bulb and effective temperature given will be observations at the control station, 30 in. from the floor.

The same general arrangement of the instruments used, the number and type of the male subjects, and the clothing worn by them, and the length of the test period as reported earlier were continued in this study. Two tests, Nos. 22 and 29, were made with female subjects in evening dress.

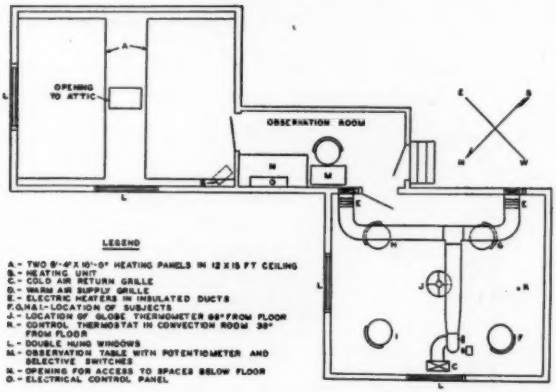
OPENING TO ATTIC
OBSERVATION ROOM
LEGEND
A - TWO 6'-6" X 10'-0" HEATING PANELS IN 12 X 15 FT CEILING
B - HEATING UNIT
C - COLD AIR RETURN GRILLE
D - WARM AIR SUPPLY GRILLE
E - ELECTRIC HEATERS IN INSULATED DUCTS
F,G,H,I - LOCATION OF SUBJECTS
J - LOCATION OF GLOBE THERMOMETER 68" FROM FLOOR
K - CONTROL THERMOSTAT IN CONVECTION ROOM 36" FROM FLOOR
L - DOUBLE HUNG WINDOWS
M - OBSERVATION TABLE WITH POTENTIOMETER AND SELECTIVE SWITCHES
N - OPENING FOR ACCESS TO SPACES BELOW FLOOR
O - ELECTRICAL CONTROL PANEL

FIG. 2. FLOOR PLAN OF TWO 12 FT X 15 FT TEST ROOMS AND OBSERVATION ROOM. DUPLICATE PROVISIONS FOR LOCATION OF SUBJECTS AND MEASUREMENT OF ALL ENVIRONMENTAL CONDITIONS

TEST OPERATION

The dry-bulb temperatures of both the panel-heated and convection-heated rooms were kept constant within small limits at or near the desired test temperature level for at least 12 hours prior to the beginning of any test. The slower response of the room temperature to the heat input to the panels made it desirable to alter in some respects the earlier test procedure. It was found that approximately ¾ hour elapsed between a change in the electrical heat input to the panel system and the response of the room air temperature. This made it more difficult to exercise the desired control of the room air temperature with changing outside air temperature and solar radiation. Consequently the tests were conducted in the early morning hours, usually between 3:00 and 8:00 A.M.

The subjects reported and entered the test room for a preliminary period of approximately 1 hour at about 3:00 A.M. During this period the room was controlled according to a previously determined schedule. Since it was desirable to have a slowly changing rather than a constant temperature during

any given test, in order to give some range in the feeling of warmth of the subjects, and since the high heat capacity of the panel made it difficult to change at will from a constant to a rising or falling temperature, and further, since the greater heat capacity of the panel made it difficult to control the temperature on a falling time-temperature curve, most of the tests were made with a rising temperature throughout the preliminary and 4-hour test period. The dry-bulb temperature was controlled to maintain as nearly as possible the previously agreed upon effective temperature schedule in the two rooms.

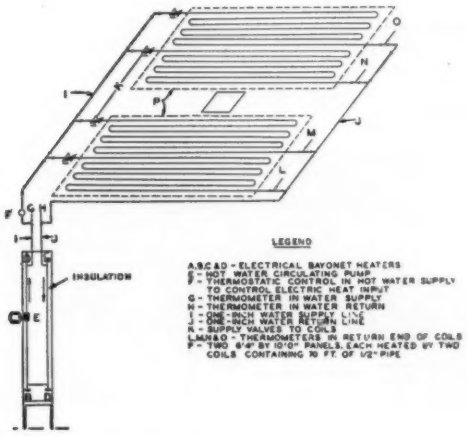
INSULATION
LEGEND
A,B,C & D - ELECTRICAL BAYONET HEATERS
F - HOT WATER CIRCULATING PUMP
G - THERMOSTATIC CONTROL IN HOT WATER SUPPLY TO CONTROL ELECTRIC HEAT INPUT
H - THERMOMETER IN WATER SUPPLY
I - THERMOMETER IN WATER RETURN
J - ONE-INCH WATER SUPPLY LINE
K - ONE-INCH WATER RETURN LINE
L - SUPPLY VALVES TO COILS
L,M,N & O - THERMOMETERS IN RETURN END OF COILS
P - TWO 8'-6" BY 10'-0" PANELS, EACH HEATED BY TWO COILS CONTAINING 70 FT OF 1/2" PIPE

FIG. 3. HOT WATER SYSTEM IN PANEL ROOM

Four of the subjects spent the first 80 min of the 4-hour test period in the panel-heated room, while the other four were in the convection-heated room; these two groups then exchanged rooms for the next 80 min and then returned to their original rooms for the last 80 min. During this period a record was kept of the electrical heat input, the temperatures throughout the rooms, and the reactions of the subjects, both as regards feeling of warmth and any unusual reactions, such as local feeling of coolness, drafts, and other unusual sensations.

TEST RESULTS

The results of each test were plotted in the analysis of the test data, similar to that in Fig. 5, which is for test No. 14 on March 12, when the outside air temperature reached a minimum of 29.6 F at 6:35 A.M. The mean radiant temperature, the dry-bulb temperature, the wet-bulb temperature, and the resulting effective temperature at the 30 in. control point are plotted for each room. The heat equivalent of the electrical input, the heat loss through the ceiling for each room, the storage (positive or negative) of heat in the *panel-ceiling assembly* for the panel heated room, and the useful heat input for each room

are also plotted. The useful heat input was obtained by deducting the heat storage and the ceiling loss from the heat equivalent of the electrical input. It will be noted from Fig. 5 that with the rising temperature during test No. 14 the amount of heat stored in the panel, ceiling and water circulating system was a considerable part of the total heat supplied. This was determined with a fair degree of accuracy by obtaining the product of the integrated temperature rise and heat capacity of the different members of the panel-ceiling assembly. Integration of the temperature was made possible by several thermocouples placed in the ceiling structure.

The average comfort votes for the four subjects in the two rooms are plotted in Fig. 5, in accordance with an index whereby 2 is indicated as cool; 3 comfortably cool; 4 ideally comfortable; 5 comfortably warm; and 6 warm. These

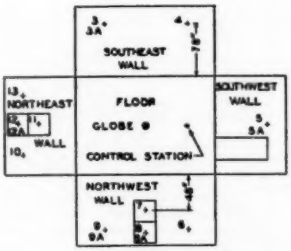

FIG. 4. LOCATION OF THERMOCOUPLES IN EAST ROOM. ARRANGEMENT IN WEST ROOM IDENTICAL, WITH EXCEPTION OF DOOR

votes are replotted against effective temperature unaffected by radiation in Fig. 6 for the panel and convection rooms, *X* being used for the subjects spending their first 80 min in the panel room, and circles for those beginning in the convection room. Curves through these points indicate the time and the effective temperature at the 30 in. level when the subjects were ideally comfortable in each room. The broken line curve for the convection-heated room is superimposed on the chart for the panel-heated room for comparison. It will be noted that in this test, the subjects in the panel room were ideally comfortable at 68.9 deg ET, while in the convection room, comfort was not had until 70.4 deg ET was reached. This difference was, of course, accounted for by the approximately 1.4 deg higher mean radiant temperature in the panel room.

Pertinent data collected in the 23 successful tests are given in Tables 1 and 2. The relation between the effective temperature at which the subjects were found to be comfortable in the different tests and the elevation of the mean radiant temperature above the dry-bulb temperature, expressed either positive or negative, is plotted in Fig. 7. Triangles and squares indicate the individual test points for the subjects in the panel and convection rooms, respectively. For comparison the individual test points from last year's study are also

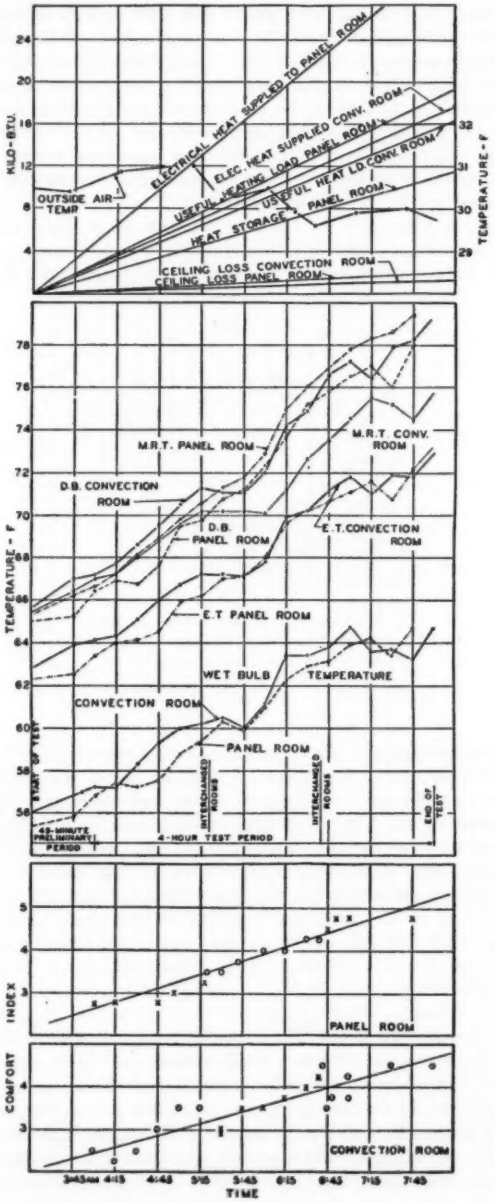

FIG. 5. LOG OF TEST 14 MARCH 12, 1941

given. This chart serves to indicate the difference in mean radiant temperature realized in the two studies. Whereas in last year's study, even in the radiator room, the dry-bulb-mean radiant temperature difference never reached a value below plus 0.9 deg, in this year's study a complete reversal was had so that in all panel room tests negative dry-bulb-mean radiant temperature differences were had. The lower positive dry-bulb-mean radiant temperature difference found in the convection room this year may be explained by the insulation applied to the ceiling of this room.

While there is a considerable scattering of the data plotted in Fig. 7, the curves giving the relation between the effective temperature and the mean radiant temperature are fairly consistent and conclusive. One outstanding fact of great importance and difficult of explanation is the approximately

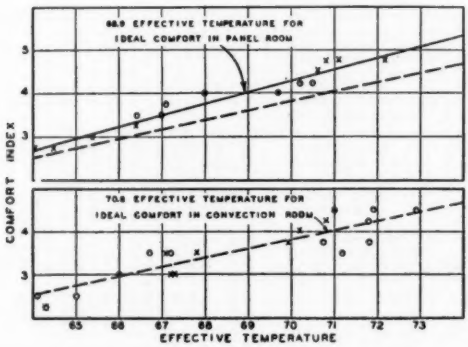

FIG. 6. RELATION BETWEEN EFFECTIVE TEMPERATURE AND FEELING OF COMFORT IN PANEL AND CONVECTION HEATED ROOMS FOR TEST 14 MARCH 12, 1941

1 deg higher effective temperature desired by the average subject in this year's test for any given dry-bulb-mean radiant temperature difference. There are several possible explanations. *First*, the fact that practically all of this year's tests were made with elevating effective temperature, while in last year's study approximately one-half of the tests were made with rising and one-half with falling temperatures, might suggest a lag in the expressed feeling of warmth. However, no such consistent differences could be found in the results for ascending and descending temperature. *Second*, a higher effective temperature might possibly be required because of some change in the prevailing habits and clothing of the subjects. However, this is hardly likely. Clothing should be a negligible factor, since in both studies all subjects were required to dress approximately alike. *Third*, it is more probable that the test in this year's study being made during the dark early morning hours from 3:00 to 8:00 A.M. may account for this difference. While no record was found in the literature of a controlled research observation of a difference in day and night heating requirement, there seems to be a prevailing opinion and practice among per-

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TABLE 1—DATA FROM 23 TESTS IN THE TWO ROOMS

		CONVECTION ROOM											PANEL HEATED ROOM									
				TEMPERATURES WHEN SUBJECTS WERE COMFORTABLE					HEAT—BTU PER HOUR				TEMPERATURES WHEN SUBJECTS WERE COMFORTABLE					HEAT—BTU PER HOUR				
TEST NUMBER	DATE	OUTSIDE AIR TEMPERATURE—F	OUTSIDE AIR VELOCITY MPH	EFFECTIVE TEMPERATURE	MEAN RADIANT TEMPERATURE	DRY-BULB MINUS MEAN RADIANT	DEGREES ET CHANGE FOR ONE COMFORT INDEX CHANGE	AIR VELOCITY AT GLOBE FEET PER MINUTE	HEAT SUPPLIED	CEILING LOSS	USEFUL	USEFUL PER DEG TEMP. DIFFERENCE INSIDE AIR TO OUTSIDE AIR	EFFECTIVE TEMPERATURE	MEAN RADIANT TEMPERATURE	DRY-BULB MINUS MEAN RADIANT TEMPERATURE	DEGREES ET CHANGE FOR ONE COMFORT INDEX CHANGE	AIR VELOCITY AT GLOBE FEET PER MINUTE	HEAT SUPPLIED	CEILING LOSS	STORAGE	USEFUL	USEFUL PER DEG TEMP. DIFFERENCE INSIDE AIR TO OUTSIDE AIR	PANEL USEFUL HEAT INPUT IN PER CENT OF CONVECTION USEFUL HEAT INPUT
6	2–28	20.1	11.0	70.3	70.2	4.6	2.4	15	4230	229	4001	73.5	67.8	73.0	−1.4	3.3	11	3732	482	−1425	4675	90.3	116.8
7	3–1	25.2	12.4	70.4	70.8	4.2	2.3	20	3754	48	3706	75.8	68.4	73.0	−1.4	3.5	17	6143	246	2275	3622	76.4	97.7
8	3–5	18.4	10.2	...	...	...	...	...	...	...	...	...	68.8	73.8	−0.6	3.4	12	7314	736	2214	4364	79.6	...
9	3–6	23.0	4.6	68.7	73.2	0.5	3.0	18	4081	400	3681	72.9	69.1	73.9	−1.6	3.4	12	6516	570	1507	4439	89.7	120.6
10	3–7	27.4	6.2	68.3	70.8	1.6	3.9	17	4041	317	3724	82.9	69.0	74.2	−0.8	3.0	15	6445	442	1572	4431	96.1	119.0
11	3–8	20.7	11.6	70.1	69.8	4.9	3.1	18	4034	246	3788	70.0	68.8	74.2	−1.2	3.1	12	6624	402	1474	4748	90.9	125.3
12	3–10	14.0	2.8	69.8	70.8	3.5	3.3	17	4777	282	4495	74.0	68.5	73.4	−1.0	3.1	11	7933	504	1638	5791	99.7	128.8
13	3–11	35.3	17.4	70.3	73.9	1.9	3.3	20	2831	202	2629	63.6	69.5	74.5	+.01	2.9	17	4978	352	1408	3218	83.6	122.3
14	3–12	30.0	22.2	70.8	73.7	2.3	4.7	16	3550	255	3295	72.8	68.9	74.0	−1.0	3.8	11	6588	400	2325	3863	90.5	117.2
15	3–13	27.1	9.2	70.1	72.3	0.1	3.0	14	3629	282	3347	71.2	68.5	72.7	−1.5	2.9	12	6760	449	2555	3756	88.4	112.2
16	3–14	23.9	5.2	70.3	74.2	0.5	3.3	18	3826	366	3460	69.1	67.7	73.2	−1.2	2.2	12	7998	628	1998	4472	91.6	129.2
17	3–15	27.2	3.2	70.2	77.1	0.2	3.5	19	3529	387	3142	63.1	69.5	76.5	−1.4	2.6	16	6232	593	1834	3805	78.8	121.0
18	3–18	8.1	22.0	70.9	69.0	5.3	2.4	16	6500	414	6086	92.1	68.7	74.3	−1.6	2.9	13	9955	787	3165	6003	92.8	98.6
19	3–19	16.8	11.4	70.5	72.0	2.0	2.0	13	5779	352	5427	94.3	68.6	73.2	−0.4	3.0	13	9320	718	3400	5202	93.2	95.8
20	3–20	33.3	3.2	68.7	75.0	0.2	2.7	11	3392	290	3632	86.6	68.5	73.3	−0.2	3.0	12	5661	420	1900	3341	83.7	92.0
21	3–21	29.3	7.4	69.2	72.5	1.5	2.2	22	4131	305	3826	84.4	69.0	74.4	−0.5	2.9	15	6831	589	2293	3949	89.5	103.2
22	3–23	24.8	14.8	69.3	72.1	1.8	2.6	20	3800	275	3525	78.5	68.0	74.8	−1.8	2.7	11	6340	500	1939	3901	88.6	110.6
23	3–24	42.7	5.2	69.5	72.1	1.0	3.9	14	2725	194	2531	87.7	69.2	73.4	−0.6	2.4	14	4334	366	1352	2616	87.2	103.2
24	3–25	31.0	14.8	70.0	73.1	1.7	2.7	17	4500	199	4301	96.6	68.4	73.6	−0.9	2.6	13	6548	473	1759	4316	105.0	100.2
25	3–26	29.4	5.0	69.2	69.3	2.3	2.4	12	...	...	...	...	69.8	73.2	−0.8	2.9	13	...	...	...	...	...	...
27	3–28	34.6	4.8	69.9	75.3	0.7	3.2	15	3425	273	3152	75.9	69.6	74.9	−0.6	3.0	18	6400	550	2029	3821	96.2	121.2
28	3–29	31.1	9.6	69.5	72.0	2.0	1.5	11	3442	352	3090	68.7	69.4	76.2	−0.2	1.1	17	2150	555	−1277	2872	63.8	92.9
29	3–30	17.0	9.6	69.8	72.0	2.2	2.7	19	4995	395	4600	80.4	68.8	73.8	−0.8	2.5	15	8500	650	2850	5000	89.2	108.7
Average		25.7	9.7	69.8	72.3	2.0	2.9	16	4046	289	3783	78.2	68.8	73.9	−0.9	2.9	14	6514	519	1763	4191	88.4	111.3

TABLE 2—DATA FROM TESTS THROUGHOUT THE TWO ROOMS AT A TIME DURING THE DIFFERENT TESTS WHEN THE SUBJECTS

TABLE 2. TEMPERATURES OBSERVED THROUGHOUT THE TWO ROOMS AT A TIME DURING THE DIFFERENT TESTS WHEN THE SUBJECTS AVERAGED COMFORTABLE

Type of Heating	No. Coils Used	Test No.	Date	Outside Air Temperature	Time of Observation	Central Control						Southeast Wall			Southwest Wall		Northwest Wall						Northeast Wall				
						1	2A	2B	2C	2D	2E	3	3A	4	5	5A	6	7	8	8A	9	9A	10	11	12	12A	13
						102″	96″	78″	30″	6″	0″	78″	78″	78″	78″	78″	78″	48″	78″	78″	78″	78″	78″	48″	78″	78″	78″
						X	Y	Y	Y	Y	X	X	Y	X	X	Y	X	G	G	Y_G	X	Y	X	G	G	Y_G	X
Convection	4	6	2–28	20.4	3:30	72.2	75.2	74.5	74.8	74.4	71.4	68.7	73.6	69.5	68.8	73.5	68.5	47.2	47.0	66.6	67.9	73.8	68.0	47.4	47.1	67.8	67.8
	4	7	3–1	26.1	3:15	72.2	75.1	75.1	75.0	73.6	69.6	69.1	73.6	69.4	69.4	74.5	68.5	49.4	49.1	67.8	68.2	74.4	67.9	49.1	48.2	67.2	68.6
	4	8	3–5	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
	2	9	3–6	23.2	6:10	73.0	74.0	74.1	73.7	72.7	70.4	68.0	72.5	68.6	68.4	73.6	67.6	48.4	48.1	67.1	67.4	73.9	67.6	48.7	48.6	67.2	67.8
	2	10	3–7	27.4	5:25	70.6	73.3	73.0	72.4	70.7	69.3	68.0	71.4	68.2	68.2	71.6	69.1	49.6	49.1	65.0	68.6	70.9	69.0	49.3	49.1	64.6	68.5
	2	11	3–8	20.6	5:11	72.0	76.6	76.1	74.7	71.8	67.7	67.4	73.2	67.8	66.2	74.9	67.0	46.7	46.0	64.9	67.7	75.7	67.2	46.6	46.4	64.7	66.8
	2	12	3–10	13.6	5:41	72.4	73.5	75.8	74.3	71.5	70.5	65.8	73.8	66.0	67.0	74.9	65.7	45.0	44.7	62.7	66.0	75.0	65.8	44.2	43.9	62.9	65.4
	2	13	3–11	34.5	6:08	73.9	76.9	76.6	75.8	73.9	73.1	70.7	74.5	70.6	70.3	76.1	70.8	57.0	54.7	69.8	70.6	76.4	70.4	56.2	55.1	69.7	71.0
	2	14	3–12	29.7	6:48	72.1	76.2	75.8	74.9	71.6	67.4	69.0	73.8	68.8	68.2	74.7	68.9	52.4	52.0	67.0	68.7	75.0	68.5	52.7	52.1	67.3	68.9
	2	15	3–13	25.4	6:09	71.8	72.6	72.8	72.4	71.2	70.9	67.6	72.0	67.7	68.1	73.0	68.0	50.3	48.3	64.4	67.9	72.5	67.2	49.0	48.5	65.2	67.5
	2	16	3–14	24.6	7:06	70.9	75.6	75.4	74.7	73.3	72.3	67.8	74.5	67.6	67.5	75.2	67.1	49.6	49.0	65.8	67.3	73.4	66.8	47.8	47.7	66.4	67.2
	2	17	3–15	27.5	6:40	75.7	78.5	78.3	77.3	77.1	73.7	70.2	75.8	72.4	71.1	77.5	68.6	52.6	51.3	71.4	68.5	76.3	71.8	51.1	49.7	70.7	72.1
	2–4	18	3–18	8.2	7:00	70.4	74.1	74.2	74.3	71.0	68.0	65.0	73.1	65.5	64.2	73.0	64.4	40.8	40.0	61.5	64.9	73.5	64.1	40.2	39.2	60.4	65.1
	2–4	19	3–19	16.5	5:48	71.3	74.6	74.3	74.0	72.8	69.8	66.1	72.9	66.8	67.4	72.0	66.7	46.2	45.0	66.3	66.6	74.2	67.3	45.4	45.0	66.2	66.3
	2	20	3–20	33.3	5:15	72.2	76.1	76.1	75.2	74.7	70.7	69.7	73.8	69.6	68.4	74.2	69.6	52.6	49.7	64.6	69.3	73.5	69.0	54.4	48.6	63.5	68.9
	2	21	3–21	28.7	5:18	72.3	74.8	74.1	74.0	73.0	70.3	69.4	73.1	69.9	69.8	75.7	69.0	51.6	50.8	68.0	69.7	73.5	70.1	52.6	50.4	68.7	70.5
	2	22	3–23	29.0	6:45	70.8	75.2	74.7	73.9	71.4	69.9	68.6	70.7	68.0	68.6	72.6	68.7	50.7	49.1	62.5	68.4	72.2	69.0	50.6	49.4	62.4	68.7
	4	23	3–24	42.5	5:24	72.0	74.0	73.6	73.1	72.4	71.5	71.0	73.9	71.7	71.2	74.8	71.5	58.5	57.9	67.2	71.3	74.3	69.2	59.3	57.4	67.0	69.5
	4	24	3–25	30.3	5:41	72.9	75.8	76.3	74.8	72.0	71.0	68.5	73.6	68.4	69.5	74.0	68.4	53.0	52.0	66.1	68.8	74.9	68.6	52.0	51.2	67.0	69.2
	4	25	3–26	29.4	6:00	71.0	75.3	74.9	71.6	70.6	69.0	68.8	70.9	68.1	68.8	72.8	68.9	50.9	49.3	62.7	68.5	72.4	69.1	50.8	49.3	62.5	68.9
	2	27	3–28	34.5	6:35	74.4	76.4	76.5	76.0	75.6	74.7	70.7	74.4	71.3	70.8	75.3	70.6	54.4	53.4	68.9	70.3	75.7	70.8	54.9	54.9	69.3	71.2
	2	28	3–29	31.1	6:54	71.2	74.3	74.1	74.0	72.6	69.7	69.2	73.0	68.8	69.1	73.9	69.0	52.7	51.8	65.8	68.4	73.2	69.0	52.9	52.5	66.0	68.2
	2	29	3–30	17.0	6:25	71.2	74.5	74.3	74.2	72.7	69.7	66.0	72.8	66.7	67.3	72.0	66.8	46.2	45.1	66.2	66.5	74.2	67.2	45.6	45.8	66.1	66.2
Panel		6	2–28	19.8	2:30	77.6	76.1	75.2	71.6	71.0	73.4	70.4	73.8	69.9	69.0	73.5	68.6	47.0	47.5	68.0	68.0	74.1	68.1	47.1	47.3	68.5	67.8
		7	3–1	24.2	3:09	77.5	75.7	73.2	71.6	69.8	71.9	71.1	73.9	71.3	69.5	74.6	69.6	49.1	49.9	66.8	68.8	78.7	69.5	49.2	48.3	67.3	69.3
		8	3–5	18.4	5:53	88.8	78.8	76.9	73.2	71.2	73.4	70.2	74.5	70.4	69.3	76.7	68.7	46.0	48.6	66.2	68.7	77.2	68.1	45.7	48.2	65.1	69.2
		9	3–6	22.8	6:22	82.4	77.2	75.6	72.3	71.0	72.6	69.8	73.0	69.8	68.6	74.7	69.4	48.5	49.6	67.2	68.3	74.0	68.5	49.0	49.3	67.4	68.0
		10	3–7	27.3	5:58	84.4	78.1	76.6	73.4	71.5	72.8	69.9	74.0	69.9	69.6	75.5	70.2	50.2	50.8	67.2	69.4	76.2	69.2	49.3	49.5	68.5	69.1
		11	3–8	20.8	5:52	98.8	80.2	77.8	73.0	70.8	73.4	71.0	74.2	70.0	70.8	76.9	69.5	46.6	48.2	65.4	69.9	77.2	68.5	47.3	48.8	64.9	68.7
		12	3–10	14.3	5:28	102.5	80.1	77.2	72.4	70.4	73.6	70.4	74.5	69.7	69.8	77.2	68.9	45.0	45.7	67.3	68.3	77.0	68.0	44.5	45.4	66.3	66.0
		13	3–11	36.1	6:00	87.3	80.6	78.0	74.6	73.5	74.0	72.6	75.3	72.1	74.3	78.8	72.5	56.1	56.9	69.3	72.4	78.9	72.7	56.6	56.7	69.4	72.4
		14	3–12	30.3	6:08	97.0	79.6	77.6	73.0	72.0	73.3	70.3	75.1	70.0	69.8	75.7	69.6	52.3	53.7	67.7	68.7	75.2	69.1	53.0	53.5	68.8	69.3
		15	3–13	28.7	5:46	88.3	75.6	74.1	71.2	69.8	71.6	70.4	72.1	71.0	68.2	74.3	71.4	51.3	49.8	64.9	71.2	73.7	68.1	49.6	49.4	65.0	68.7
		16	3–14	23.2	6:08	91.6	76.7	73.1	72.0	71.2	72.6	70.8	74.5	68.9	67.6	75.2	68.5	51.3	49.1	65.7	69.6	74.8	68.9	47.4	48.0	68.9	68.9
		17	3–15	26.8	6:27	99.1	82.4	80.0	75.1	73.6	75.2	74.7	76.7	75.2	73.1	79.0	68.7	54.6	52.5	71.2	68.6	79.8	72.8	50.8	50.3	74.7	72.9
		18	3–18	8.0	6:34	107.2	84.0	80.5	72.7	70.0	71.5	69.0	76.5	70.1	69.0	80.4	68.0	41.0	41.9	68.0	67.8	77.0	69.8	40.0	40.6	69.3	69.0
		19	3–19	17.0	5:14	100.4	81.3	77.5	72.8	69.9	71.8	69.2	74.8	69.5	70.4	78.4	68.7	46.9	45.1	66.9	68.9	78.9	70.5	45.8	46.8	67.6	70.4
		20	3–20	33.2	5:15	83.2	77.2	75.1	73.1	71.6	73.2	71.2	74.9	71.5	68.9	75.2	71.1	53.3	51.2	65.9	69.6	76.6	70.6	54.0	52.0	70.7	69.8
		21	3–21	29.8	5:06	81.8	76.7	75.8	73.9	72.1	72.7	71.7	77.0	71.5	70.2	76.4	70.6	53.3	51.4	68.3	69.8	77.5	70.5	52.2	52.1	69.8	70.8
		22	3–23	29.0	6:00	83.0	76.2	74.9	73.0	70.9	72.0	70.2	72.1	69.7	69.4	74.7	70.3	50.3	49.5	68.0	71.2	74.1	70.7	50.4	50.5	69.8	70.3
		23	3–24	42.8	5:08	77.4	75.9	74.7	72.8	72.2	71.7	74.5	76.4	74.8	74.5	75.6	74.0	59.7	58.7	68.0	73.6	75.7	73.0	59.3	59.1	71.4	73.2
		24	3–25	31.6	5:11	82.7	77.4	74.9	72.7	71.5	72.7	70.5	73.6	70.8	71.0	75.8	70.8	53.3	52.4	68.1	69.5	76.3	70.3	51.0	51.6	71.8	70.3
		25	3–26	29.3	6:06	80.9	76.4	75.1	72.4	71.1	72.1	70.4	72.3	69.9	69.5	74.9	70.4	50.5	49.4	68.2	71.3	74.3	70.8	50.5	50.2	70.0	70.4
		27	3–28	34.6	6:41	78.9	77.2	76.1	74.3	73.5	75.6	72.3	74.7	72.2	71.0	75.4	71.4	54.5	55.0	69.2	70.6	75.7	71.1	55.0	55.5	72.8	71.7
		28	3–29	31.0	6:48	86.1	79.7	78.5	76.0	75.7	77.5	71.7	76.2	71.9	72.7	77.2	71.0	53.7	53.8	72.2	71.4	77.4	71.8	54.0	54.6	76.1	71.8
		29	3–30	17.0	6:10	98.6	80.2	78.1	73.0	70.2	71.9	69.1	74.7	69.5	70.3	78.3	68.6	46.8	46.2	66.8	68.8	78.7	70.4	46.0	46.7	67.5	70.3

Location of observation points as indicated in Fig. 4. Distances measured from floor. Therefore: 0 in. = Floor surface. 102 in. = Ceiling.
X = Surface temperature Y = Air temperature. G = Glass surface temperature. Y_G = Air 6 in. from glass surface.

sons interested in the heating and temperature control of industrial plants employing both day and night workers, indicating such a difference. The tests analyzed give a very interesting demonstration of the actual existence of this fact. The difference between the day and night heating requirements may possibly be accounted for by certain radiation effects, or more plausibly, by a lowered vitality and metabolism of persons during these hours.

Figs. 8 and 9 give interesting relationships between the mean radiant temperature, the dry-bulb-mean radiant temperature difference, and the dry-bulb

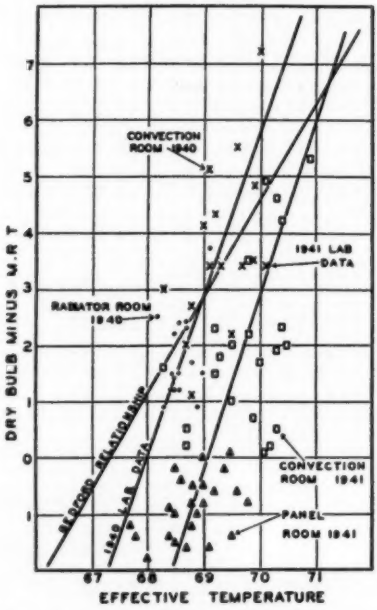

FIG. 7. RELATION BETWEEN EFFECTIVE TEMPERATURE UNAFFECTED BY RADIATION AT WHICH COMFORT WAS REALIZED AND DIFFERENCE BETWEEN DB AND MRT

temperature in the two rooms, and the outside dry-bulb temperature at the time when the subjects were comfortable. The single point on each curve at slightly under 74 F outside temperature was determined in connection with last year's study as representing an equilibrium condition, when the mean radiant temperature and the dry-bulb temperature of the atmosphere were all the same with 68 deg ET and 40 per cent relative humidity. This point was established by tests in the Laboratory's psychrometric rooms for winter conditions and reported in last year's study. This point serves admirably in fixing the shape of the curves.

The same tendency for the mean radiant temperature to compensate for falling outside temperature as was experienced in the radiation and convection rooms last year is indicated for the panel and convection rooms this year; even to a slightly greater extent. In the convection room, where radiation is a lesser factor, the mean radiant temperature goes down progressively with falling outside temperature, due to the colder inside surface of the outside walls and windows. The slightly lesser downward trend in this year's convection room results undoubtedly comes from the application of insulation

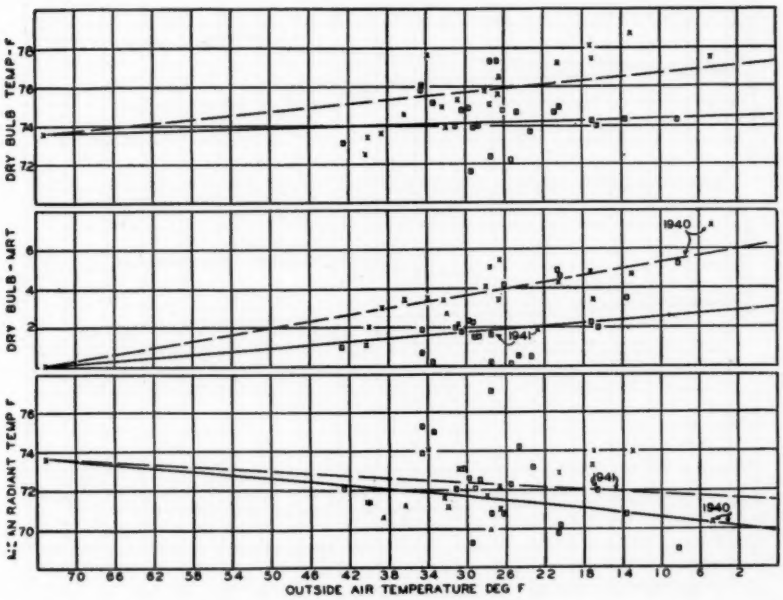

FIG. 8. RELATION BETWEEN OS DRY-BULB TEMPERATURE AND MRT, DRY-BULB MRT AND DRY-BULB FOR CONVECTION ROOM

above the ceiling. In the panel-heated room the necessary elevation in the panel surface temperature in order to provide the increasing amount of heat with falling outside temperature tends to offset the falling inside surface temperature of the outside walls, so as to keep the mean radiant temperature constant for the entire series of tests. It should be also emphasized that not only does the elevated temperature of the panel surface serve to raise the mean radiant temperature, but that also, the increased radiation from the panel raises somewhat the temperature of all surfaces in view of the panel, including the inside surfaces of outside walls and glass, the floor, the occupants' clothing, the furniture and the furnishings. This will be clearly indicated in a comparison of these surface temperatures in the tabulated data.

The effect of the elevated mean radiant temperature in the panel heated room with falling outside temperature serves to stabilize the relationship between outside air temperature and the difference between the inside dry-bulb temperature at the central control point and the mean radiant temperature. In the convection room the value of this difference, or the depression of the mean radiant temperature below the dry-bulb temperature, increases progressively with falling outside temperature. Again the effect of the insulation of the

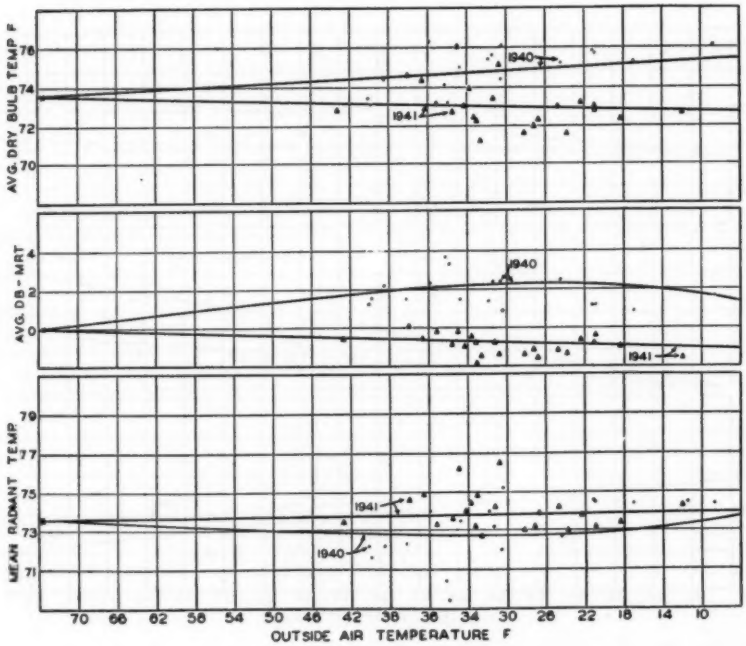

FIG. 9. RELATION BETWEEN OS DRY-BULB AND MRT, DRY-BULB MRT AND DRY-BULB FOR PANEL ROOM

ceiling is shown by the lower slope of the curve for this year's data. In the panel-heated room most of these values become negative; that is, the mean radiant temperature was always equal to or greater than the dry-bulb temperature unaffected by radiation at the control point at the time of comfort.

The dry-bulb temperature at the control point in the convection room required elevation with falling outside temperature in order to compensate for the increased radiation effect from the subjects to surrounding surfaces resulting from the lower mean radiant temperature. Again the effect in the convection-heated room this year is decreased; in fact, almost to the extent of disappearance, due to the added insulation, while in the panel heated room, the elevation of

the mean radiant temperature above the dry-bulb temperature gave sufficient compensation so that with lower outside temperatures there seems to be some tendency for a lower rather than a higher dry-bulb temperature to be required. In this connection, it may be well to indicate that another disturbing factor serves to lower slightly the dry-bulb temperature in this year's study, namely, the relative humidity maintained within the rooms. As pointed out above, the relative humidity could only be controlled within certain limits. It was felt desirable to maintain somewhat lower relative humidities for the colder outside temperatures in order to avoid condensation on the windows, since this disturbed the window condition as well as giving a false indication of the heating requirement. As an average, slightly higher relative humidities were main-

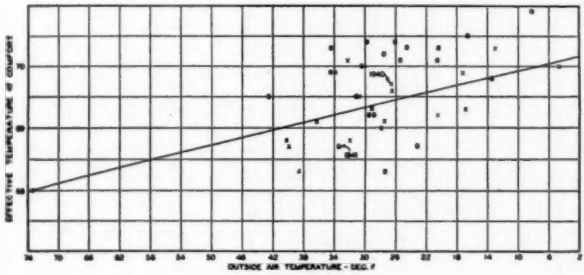

FIG. 10. RELATION BETWEEN EFFECTIVE TEMPERATURE AT WHICH COMFORT WAS REALIZED AND CONCURRENT OUTSIDE AIR TEMPERATURE FOR CONVECTION ROOM

tained in this year's tests, necessitating a lower dry-bulb temperature for the same effective temperature.

Figs. 10 and 11 give the relation between the outside temperature and the effective temperature required at the 30 in. control point for the two rooms. While in the convection room, a progressively higher effective temperature was required for comfort with lowered outside dry-bulb temperature, in the panel room there was a tendency to compensate so that relatively little elevation in the effective temperature is indicated for cold outside temperatures, even compared with the single control point at 68 deg ET when radiation was not a factor.

While the purpose of this investigation was essentially to study the effect of radiation on the feeling of warmth, as compared with the temperature, moisture content and velocity of the air, and while the types of heating systems used were essentially to give variation in the radiation effect, the great interest in these respective systems made it desirable to attempt to get some figures on the relative heating requirement for the three. Last year's investigation showed that on the average the radiator system required approximately 10 per cent more heat to maintain comfort that was the case in the convection system.

Fig. 12 shows the relation between the electric heating requirements of the panel and convection rooms. The location of the panel in the ceiling, and the

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requirement that insulation be placed above it in order to conserve heat, made it desirable to also insulate the ceiling of the convection room in order to give a better comparison. Nicholls' heat flow meters placed on top of the insulation above the ceiling in both rooms made it possible to evaluate the heat escaping through the ceiling in the two instances. In order to make the comparisons of the actual heat requirements within the rooms themselves comparable, the ceiling loss was deducted from the heat supplied to each room. Since most of the tests in this year's study were made with elevating temperature throughout the test period considerable heat was stored within the panel-ceiling assembly and in the panel heating system during the test. As previously pointed out, it was possible to integrate the temperature rise—heat

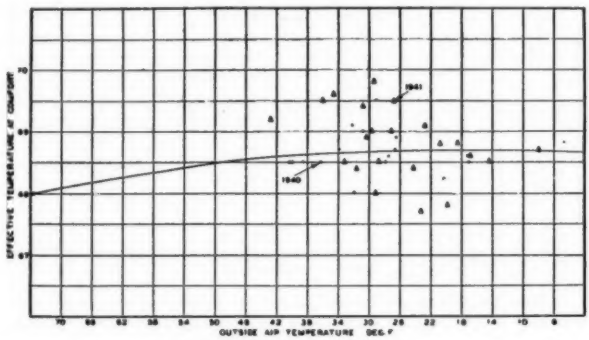

FIG. 11. RELATION BETWEEN EFFECTIVE TEMPERATURE AT WHICH COMFORT WAS REALIZED AND CONCURRENT OUTSIDE AIR TEMPERATURE FOR PANEL ROOM

capacity relationship for the entire assembly with some degree of accuracy. This made it possible in Fig. 12 to correct the heat supplied to the panel room for the storage in the panel and the ceiling loss for that room and for the ceiling loss for the convection room, and thereby to offer a comparison of what is here termed the *useful heat input* for the two rooms. This shows an increase of about 11 per cent in the average case for the panel room.

The reason for the increase in heating requirement with the increase in the mean radiant temperature is apparent. The radiation from the warm panel surface raises the temperature of all surfaces in view of it, including the inside surfaces of all outside walls (including the glass) resulting in an increase in the heat transfer to the outside. It is therefore apparent that any system which elevates the mean radiant temperature will have an increased heat loss if other factors remain the same.

The temperature gradient from the ceiling to the floor in the panel and convection rooms for this year's study, and for the radiation room in last year's study are given in Fig. 13. Curves are drawn for the average of all tests for each room at the time when the subjects were comfortable. The temperature was controlled at the 30 in. level; hence the small variation in the

level of the curves at this elevation. The small difference results from the variation in the dry-bulb temperature at comfort with the different heating methods.

It is of interest to note the higher floor surface temperature in the panel room resulting from the absorption of radiant heat as compared with that for the convection room. This should serve to give some comfort advantage for this method of heating. The higher temperature of the air approaching the panel and the lower temperature near the ceiling of the convection room are no doubt important factors in the greater heat loss shown for the panel room. The area between the panel room and the convection room curves between the

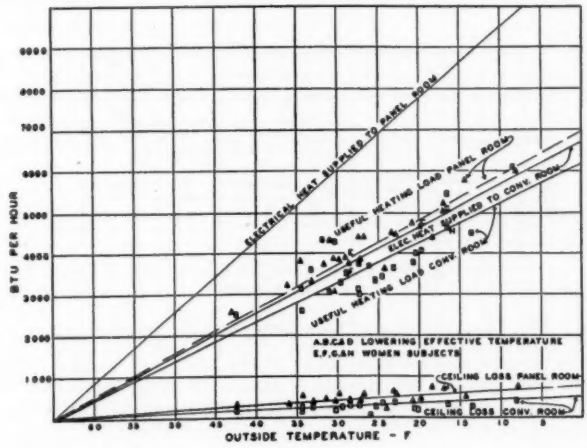
BTU PER HOUR
OUTSIDE TEMPERATURE - F
ELECTRICAL HEAT SUPPLIED TO PANEL ROOM
USEFUL HEATING LOAD PANEL ROOM
ELEC. HEAT SUPPLIED TO CONV. ROOM
USEFUL HEATING LOAD CONV. ROOM
ABCAD LOWERING EFFECTIVE TEMPERATURE
E.F.G.AN WOMEN SUBJECTS
CEILING LOSS PANEL ROOM
CEILING LOSS CONV. ROOM

FIG. 12. ELECTRICAL HEAT SUPPLIED TO HEATING UNITS, HEAT LOSS THROUGH CEILING, HEAT STORED IN WATER SYSTEM AND PANEL ROOM AND THE RESULTING USEFUL HEATING LOAD FOR THE CONVECTION AND PANEL HEATED ROOMS

floor and the approximately 4 ft level is important in the feeling of warmth for the seated person and is a measure of counteracting effect of the greater mean radiant temperature in the panel room.

DISCUSSION OF RESULTS OF THE STUDY

The paper serves in general to verify the results of last year's study as regards the relationship between the effect of mean radiant temperature, and the effective temperature of the air in determining a person's feeling of warmth. While slightly higher effective temperatures are indicated by the data plotted in Fig. 6, the same slope of the curves is indicated for the two studies. As determined by these curves a 1 deg ET fall is compensated for by a 3.13 deg elevation in the mean radiant temperature, or a 1 deg elevation in the mean radiant temperature is compensated for by a 0.32 deg ET depres-

sion. These curves as drawn from the data show some contrast with the results of the analysis of Bedford's work as contained in the last report [3] and shown by a curve in Fig. 7, where an elevation of 1 deg in the mean radiant temperature is compensated for by a depression of 0.57 deg ET.

The data in Table 1 show that an average change of 2.9 deg ET was required to change the feeling of warmth from ideally comfortable to either comfortably warm or comfortably cool. It would follow, therefore, that a change of 9.1 deg in the mean radiant temperature would be required to change the feeling of warmth from ideally comfortable to comfortably warm or comfortably cool with no other change in the environmental condition. The data plotted for both studies in Fig. 7 show an extreme range of approximately 8 deg in the difference between the dry-bulb temperature and the mean radiant

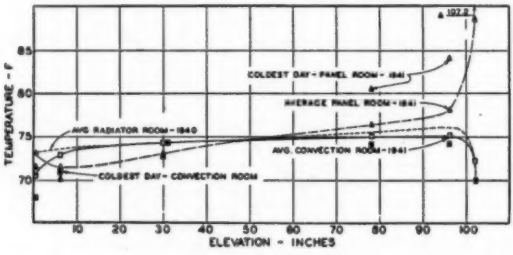

FIG. 13. CEILING TO FLOOR TEMPERATURE GRADIENT FOR AVERAGE OF ALL 1831 TESTS IN THE PANEL AND CONVECTION ROOMS WITH TEST POINTS FOR THE COLDEST DAY TEST 17, 8 F MINIMUM, ALSO AVERAGE OF ALL 1940 TESTS IN RADIATOR ROOM. TEMPERATURES AT 0 IN. AND 102 IN. ARE FLOOR AND CEILING SURFACES

temperature for all tests in both years' study. In other words, if no other environmental conditions than the mean radiant temperature had been changed in any test for the three types of systems studied, the maximum effect of the mean radiant temperature change would have been only sufficient to change the feeling of warmth by as much as from ideally comfortable to comfortably warm or comfortably cool. Thus, it will be seen that the three methods of heating involving widely different ratios of radiated and convected heat additions to the enclosures gave relatively little difference in the feeling of warmth experienced by the occupants.

The fact that the mean radiant temperature was not successfully elevated by the panel system by more than 2 deg above the dry-bulb temperature in any test is of interest. However, it should be emphasized that the current study represented a single and preliminary attempt to elevate the mean radiant temperature by the use of panel heating, and that the possibility of further elevating it was by no means exhausted. The panel used was large, and it was therefore not possible to heat its surface to a very high temperature without overheating the room. This was pre-arranged as a desirable feature;

[3] Loc. Cit. Note 2.

however, it may be possible that the mean radiant temperature could be more easily elevated by a smaller panel area and a greatly increased panel surface temperature. The accepted opinion on this point is controversial. Nevertheless, if this study is not conclusive on the success of greatly elevating the mean radiant temperature by panel heating, it at least serves a useful purpose in this direction, and further studies along this line should be helpful.

In this connection it may be well to point out the fact that reports in the literature of instances where extremely low dry-bulb temperatures were found to be comfortable with greatly elevated mean radiant temperatures do not at all simulate building heating practice. In some instances [4] the space heated was an outside porch arrangement, or a room without a side wall, and in other instances [5] excessive ventilation rates of cool air were supplied in order to make the exaggerated effect possible.

While the study seems to indicate that sufficient information is available to properly evaluate the effect of radiation in the feeling of warmth within the zone of comfort, further work must be done before this environmental factor can be completely evaluated in the effective temperature index to the same extent that the other three environmental factors are: namely, dry-bulb temperature, moisture content and air movement. It must be admitted that there will arise many instances where the mean radiant temperature is elevated far above that indicated in this study, and if such cases are construed as being economically important, then this phase of the subject should be extended to include higher mean radiant temperatures.

The present growing interest in panel heating and the indication that it is rapidly becoming of economic importance as a method of heat application, together with the experience gained in this study, emphasizes the importance of further fundamental studies of such basic factors in panel heating as the relative effect of panel size and panel surface temperature on the mean radiant temperature which may be obtained in different types of rooms. This relationship will undoubtedly be affected by such factors as the relation between heat loss by wall transmission and by ventilation and either controlled or uncontrolled infiltration.

Further study should be made of the effect of such factors as the shape, location, and temperature distribution of the panel surface, the method of applying the heat to its interior, the heat emission per square foot of panel, the allowable panel surface temperatures without adversely affecting the comfort of occupants when the panel is in the floor, side wall, or ceiling, and the effects of the panel heat capacity on the temperature control of the enclosure.

While the relatively high heat capacity of panel-ceiling assembly and heating system was found to adversely affect the desired control over temperature conditions within the panel-heated room used in this study, as previously mentioned, it should be emphasized that those disadvantages pertained largely to the experimental character of this study, and that they may not at all apply as important factors in the practical problem of space heating. In this study it was desired to change the temperature of the air up and down at will, while in heating practice the reverse is usually desired, namely, temperature stabilization.

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[4] Room Warming by Radiation, by A. H. Barker. (ASHVE TRANSACTIONS, Vol. 38, 1932.)
[5] Obtaining Comfort Conditions by Controlled Radiation from Electrically Heated Walls, by L. W. Schad. (*AIEE Transactions*, 1933, Vol. 52).

DISCUSSION

W. L. FLEISHER, New York, N. Y. (WRITTEN): This paper brings to a head my continual criticism of curves between statistical points, establishing a line of expectancy. It seems to me that in almost every case I have observed over the last 20 years, the salient facts in connection with the establishment of comfort zones and comfort lines have been lost in an attempt to interpret individual reactions as determining a group trend.

In this particular investigation a line has been drawn between the points expressing the sensations of warmth or coolness of individual subjects. At the point selected as optimum, along the so-called "4" line half the subjects were slightly cool and half were slightly warm. Not a single subject was ideally comfortable, and half had diametrically opposite sensations from the other half. If that is not interesting from a physiological and psychological angle, I cannot imagine any observations that are, but I certainly cannot see how anyone can draw from these observations the conclusions that that particular point or line or zone is the comfort zone. Maybe the subjects are not very uncomfortable, but they are not ideally comfortable and, more important than that, they have almost exactly opposite sensations. The pertinent question is, why these subjects have exactly opposite sensations and how can we measure physically the differences that develop these opposites.

It is up to us and the medical fraternity and the physicists to find out in what way these people differ fundamentally, skin area, surface moisture, normal temperatures, skin temperatures, metabolism of each of the subjects, then an orientation of his physical measurements to his sensation, and probably some great discovery.

The trouble is that we are measuring and predicting with a yard stick something we should be investigating with a microscope. When we arrive at a critical point where everyone is too cool or too warm, we have the limiting points in our integration that may make it possible to determine the discrepancies at the intermediate points.

We have a great series of observations that show that at 79 deg ET no one feels comfortable, and we have a similar point or line at 64 deg ET. Between these points we have expressions of comfort or warmth or coolness on both sides of the line. If our measurements are to be exact, we would have to have nude subjects, or we would have to know much more fully than we now do the effect of clothing or the materials of clothing. Not only that, we would have to know from individual subjects the effect of clothing on different parts of the body.

In a paper that the Fleishers presented to the Society, we tried to establish the reactions of human beings as a function of the kinetic energy coming in from the enveloping space and to show that at 99 deg, plus or minus, at 0 deg relative humidity, corresponding to the blood temperature no one was comfortable unless his normal blood temperature was above the so-called normal of 98.6 F.

I feel that we have done enough drawing of lines or bands or curves, and with the vast amount of data now available we should retrace our steps and attempt to draw conclusions only from the individual and not from the mean reaction of the group.

J. C. FITTS, New York, N. Y.: It seems to me that in this investigation, the Research Technical Advisory Committee on Radiation and Comfort has been doing some pioneering work. We did not offer this paper as a finished product, but rather as a second installment of an exploratory study. I am not entirely sure that the Society has reached the point where we can define in definite terms the effect of radiation on the environment of an individual in a room, but I think we are approaching much closer to the answer.

The work that Mr. Hutchinson has presented and the study that has been reported covering work done at the ASHVE Research Laboratory are both planned to obtain fundamental data. We realize that there are a number of factors that must still

be investigated at the Pittsburgh Laboratory and in the other cooperative institutions, but if the membership will take these two papers as another step to connect up the environment with comfort, I think you will feel that the Society Laboratory has accomplished something that is worth while and has laid the foundation for more study that will be of great value.

C.-E. A. WINSLOW, New Haven, Conn.: I think this is a most admirable study, but after all, in discussion it is the point of difference that is important. The major point of difference here is that Professor Raber and Mr. Hutchinson have found, as we have in New Haven, that under a certain set of conditions the lowering of one degree of mean radiant temperature will require a raising of one degree of air dry-bulb temperature, while Dr. Houghten's results are notably different. It is important to remember that this relation varies with the rate of air movement which must be taken into account because any increase in air movement increases the heat loss by convection and does not, of course, have any effect on the radiation characteristics of the space.

I am not quite sure from the analysis of the figures presented that the discrepancy is as great as is indicated. It is a little confusing to use effective temperature in this particular computation but I note from Dr. Houghten's tables that the equivalent comfort conditions averaged about 72.3 for radiant heating and 73.4 for the other type of heating. In each case the average has been determined from the figures presented, all of which seem to tend to substantiate the one to one comparison for mean radiant and dry-bulb temperature.

I think it would be important to analyze a little more thoroughly the curves from which the three to one conclusions have been derived. This is a matter which has been brought up before in these meetings. The points on these graphs are quite highly scattered and I believe I should have drawn the lines quite differently. I notice that Dr. Houghten said that the points did not fit the curves. I think the question to raise is whether the curves fit the points.

F. W. HUTCHINSON, Berkeley, Calif.: I think these two papers bring out the point quite often overlooked. The work at the University of California aside from the particular paper which I presented, has largely been theoretical, whereas Dr. Houghten's work at Pittsburgh has largely been in the collection of experimental data and frequently the two come very close together. Taking the result of Dr. Houghten's paper that the depression of the mean radiant temperature minus the air temperature was around 0.9 F; these check within 10 per cent of theory. I would not have made that statement a couple of hours ago, but checking with the experimental data we find that they correspond very closely with the equations.

The data from the experiment which has been reported were based upon the ventilation rate of less than one air change per hour in a room having an equivalent over-all heat transfer factor of 0.20. For example, in a particular panel-heated room of an office building, we think now that we can show that the depression is equal to a direct function of any temperature difference.

The theory indicates that the depression increases rapidly with ventilation rate and also varies as a function of the type of construction. This latter effect is evidenced by the difference between English data based on office buildings and United States data collected largely from residential type buildings. Obviously, a room in a large structure having small exposure has a much lower equivalent conductance.

The point I want to make is that Dr. Houghten's results check quite closely with theory and if this is more than a coincidence then it would indicate that the depression value could be much greater for different types of construction.

DR. HOUGHTEN: The only point that seems to require further amplification is the question raised by Dr. Winslow, regarding the points and the curves drawn through them. It is impossible to obtain a curve that will fit all of the points; this has been true in all our physiological studies. The best that can be done is to take some kind of a statistical average.

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The determination of the relationship between mean radiant temperature and the effective temperature is based on something more than on these curves for it involves some of Bedford's work. We are not altogether in agreement with Bedford's studies at high temperatures, which relate the mean radiant temperature to the dry-bulb temperature. However, by taking into account Bedford's relation between mean radiant temperature and our relation between radiant temperature and effective temperature we have been able to compare the two pieces of work. There is relatively little difference between Bedford's earlier findings in higher temperatures obtained in industrial work and the results reported in this paper.

During the last month there has been occasion to make a comparison of a little different nature, based upon physiological reactions. We have been studying the effect of hot atmospheres on the elevation of body temperature of workers, and in some of those studies the subjects were affected by considerable radiation. These tests were all made at high temperatures, ranging from 90 to 105 deg ET and in some cases with as much as 16 F elevation in mean radiant temperature above the dry-bulb. Hence, we now have data on rise of body temperature for persons in still air with and without radiant heat and the results indicate an effect due to elevation in mean radiant temperature equal to less than half the same number of degrees of elevation in effective temperature.

It was difficult in this study to obtain very accurate relationships between the relative effects of elevation in mean radiant and in effective temperature because of the inability to vary the mean radiant temperature over wide limits. The minimum radiant temperature in the room with no radiation was 6 deg below the dry-bulb, while the maximum in the panel heated room was 2 deg above the dry-bulb.

No. 1194

PERIODIC HEAT FLOW IN BUILDING WALLS DETERMINED BY ELECTRICAL ANALOGY METHOD

By VICTOR PASCHKIS,* NEW YORK, N. Y.

INTRODUCTION AND SCOPE

IN order to choose the most suitable wall from a thermal consideration and to select either the cooling or heating equipment to maintain a prescribed inside air temperature for a given wall, it is necessary to determine the time-heat flow and the time-temperature curves.

The flow of heat through building walls has been the subject of many investigations conducted either by mathematical analysis or by direct experimentation.[1]

This report describes two experiments conducted by the *electric analogy method*.[2] Both experiments cover conditions of test previously conducted in the ASHVE Research Laboratory located in Pittsburgh, Pa., by a direct thermal method.

The analogy or model method of investigating transient heat flow problems is based on the identity of the equations governing transient heat flow and the flow of electricity in a resistance-capacity circuit. There is a perfect analogy between heat and electric phenomena as compiled in the following scheme:

Temperature	Electric voltage
Heat flow (per unit time)	Electric current
Thermal capacity (specific heat multiplied by density)	Electric capacity
Thermal conductivity	Electric conductivity (reciprocal of electric resistivity)

* Research Associate, Department of Mechanical Engineering, Columbia University.

[1] RESEARCH REPORT No. 923—Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh and Paul McDermott. (ASHVE TRANSACTIONS, Vol. 38, 1932.)

Some Applications of Physics to Air-Conditioning Physics, by E. M. Pugh. (*Physics*, Vol. 7, March, 1936.)

Transient Heat Load in Calculating Air Conditioning Loads, by E. M. Pugh. (*Refrigerating Engineering*, Vol. 42, July, 1941.)

Effect of Heat Storage and Variation in Outdoor Temperature and Solar Intensity on Heat Transfer Through Walls, by J. S. Alford, J. E. Ryan and F. O. Urban. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

The Flow of Heat Through Walls, by F. E. Giesecke. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

A Laboratory Method for Cyclic Heat Measurements on Walls and Roofs, by E. R. Queer and F. G. Hechler. (ASHVE TRANSACTIONS, Vol. 47, 1941.)

[2] A Method for Determining Unsteady-State Heat Transfer by Means of an Electrical Analogy, by Victor Paschkis and H. D. Baker. (*ASME Transactions*, 1942.)

Presented at the 48th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January, 1942.

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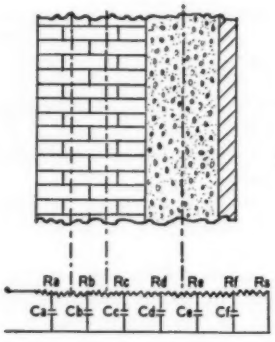
It is therefore possible to arrange an electric circuit having certain characteristics, and to compute the desired heat values from electrical measurements. A non-mathematical description of this method has previously been published [3] and it will suffice in this discussion to describe the method by one example.

Assume a wall (Fig. 1), subjected to heat flow on one side, and air of constant temperature on the other side. In order to represent the wall in the electrical model (analogue), it is considered to be cut into sections, as indicated by the dotted lines in Fig. 1. Each section is represented in the model by an electric condenser (for the thermal capacity) and electric resistors for the thermal conductivity.

After arranging the model circuit, voltage is applied; if the temperature is constant, constant voltage is applied; if the temperature changes (*e.g.*, periodically) the voltage is also changed (periodically). There is a definite ratio between temperature and voltage, based on the ratio of electric and thermal resistivities and capacities, *e.g.*, 1 F may be represented by 2.5 volts. The current is measured and can be converted into thermal values expressed in terms of Btu per hour.



The first part (*brick*) is considered to be cut into three sections; the second (*concrete*) in two; while the third layer (*plaster*) is represented by one section only. Each section is represented in the electrical model (wiring diagram under the sketch of the wall) by one resistor and one condenser. The resistors are marked *Ra*.....*Rf*; the condensers are marked *Ca*.....*Cf*. An additional resistor *Rs* represents the film conductance on one side of the wall. The other side of the wall is subjected to a given heat flow.

FIG. 1. DIAGRAM OF WALL CONSISTING OF BRICK, CONCRETE AND PLASTER

It is important to know that by appropriate choice of the electrical values, a *time ratio* can be selected, that is, a heat cycle of 24 hours can be *condensed* proportionally, to last in the model only 15 min. Or a very short heat cycle, lasting only fractions of a second, can be *stretched* to last several minutes.

EXPERIMENT A

The Problem

Given a building wall and the temperature-time curve for the outside surface. How does the heat flow from the inside surface change during one cycle, if the inside air temperature is held at a constant value?

[3] Electrical Machine Solves Heat Transfer Problems, by Philip W. Swain. (*Power*, July, 1941.)

The experiment was conducted for the conditions prevailing on July 25, 1931 for a 2-in. pine panel described in a previous paper.[4]

The physical data for this wall were as follows:

Actual thickness (including layer of plaster)—2.16 in.
Thermal conductivity—0.83 lb per square foot, per degree Fahrenheit per hour per inch.
Density—27.46 lb per cubic foot.
Specific heat—0.467 Btu per pound per degree Fahrenheit.
Inside film conductance—1.9 Btu per square foot per degree Fahrenheit per hour.

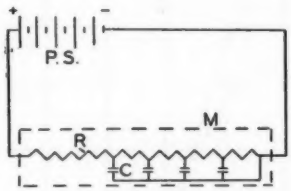


FIG. 2. CIRCUIT DIAGRAM

P. S.=power supply *R*=resistors
M=model *C*=condensers

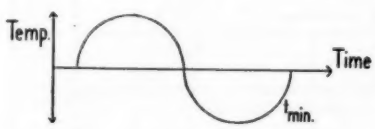


FIG. 3. TEMPERATURE-TIME WAVE

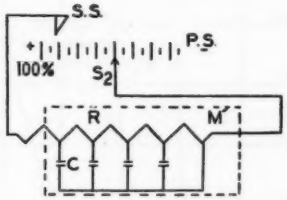


FIG. 4. CIRCUIT DIAGRAM FOR EXPERIMENT A

P. S.=power supply
S. S.=selector switch, taking any voltage from +(100%) to zero
S_2=switch taking voltage for constant room air temperature
M=model, consisting of *R* resistors, and *C* condensers.

The thermal properties and the conductance were considered to be independent of temperature.

Procedure

The general procedure is described in the previously mentioned paper,[5] but the present problem involved some special features which required some alterations in the equipment, explained later and referred to in Fig. 2.

In a great number of transient heat flow problems, the heat flows in only one direction without reversing. Such heat flow problems can be represented by an electric model circuit, Fig. 2, where the potential of one end of the circuit is chosen as zero. The circuit itself consists of resistors and condensers which are respectively the analogues of the thermal resistance and the thermal capacity. For the present case the heat flow changes its direction during each cycle and the temperature-time curve swings periodically around a zero line, *e.g.*, the constant room (air) temperature.

The available recording voltmeters have the zero line at the left side of

[4] Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh and Paul McDermott. (ASHVE TRANSACTIONS, Vol. 38, 1932.)
[5] Loc. Cit. See Note 2.

the scale, and thus do not allow the recording of voltage oscillating on both sides of zero. In order to use the instruments, the zero potential for the experiment had to be selected in such a manner that the whole oscillation of the cycle was on one side of the zero potential line. If Fig. 3 represents such a temperature-time cycle (or voltage-time cycle in the model), then the

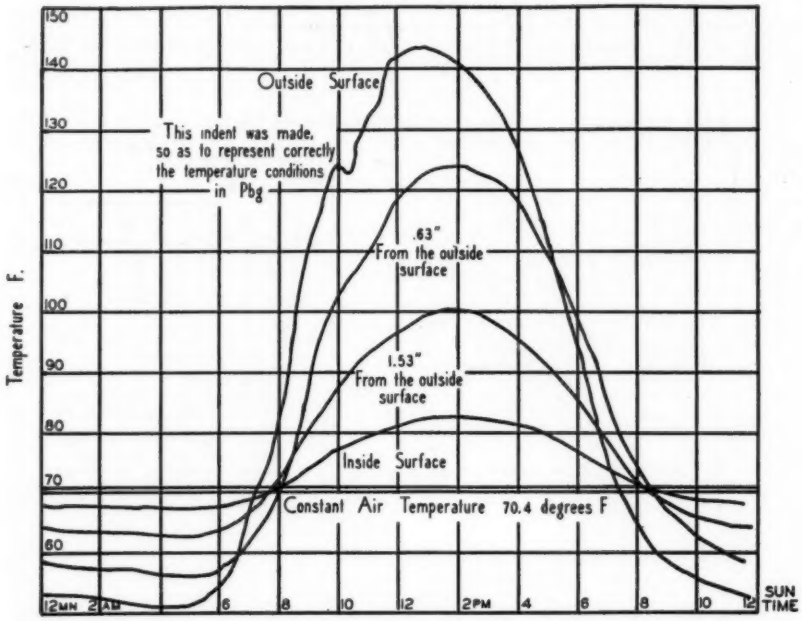


FIG. 5. TEMPERATURE-TIME CURVES; TEMPERATURES AT THE OUTSIDE AND INSIDE SURFACES OF THE WALL AND WITHIN THE WALL FOR EXPERIMENT A, 2-IN. PINE WALL

zero potential must be equal (or lower) than t_{min}. The circuit diagram of Fig. 4, conforms to this requirement.

For the experiments with the model a time ratio was chosen. In the present case a ratio of 112.5 was selected; 1 sec in the model experiments represented 112.5 sec of heat flow through the wall. The daily cycle of 24-hour was thus represented by $\frac{24 \times 3600}{112.5} = 768$ sec.

The temperature-time cycle for the outside surface as drawn in Fig. 5 showed some irregularities, and these irregularities were imitated in carrying out the *model experiments*. The voltage representing the outside surface temperature was regulated according to the changes of the outside surface temperature, as recorded in 15-min intervals by the ASHVE Research Laboratory.

Each 15-min interval was covered by two steps, representing 7.5 min each. The duration of each step in the experiment was $\frac{7.5 \times 60}{112.5} = 4$ sec, which is ample for setting the correct value.

The voltage-time cycle, representing the temperature-time cycle, was repeated seven times in order to make sure that two consecutive cycles were repetitions without any charging or discharging phenomena.

By means of recording voltmeters, the temperatures on the outside surface and on the inside surface were recorded. Furthermore, the temperature-time curves at points 0.63 in. and 1.53 in. from the outside surface were taken.

In intervals of 8 sec (corresponding to 15 min of heat flow) the current representing the heat flow entering the *outside surface* and the current representing the heat flow leaving the *inside surface* were read.

Results

The temperature-time curves are given in Fig. 5 with the time plotted as abscissa, and the temperatures as ordinates. Four curves were drawn: one for the outside surface; the next for a layer 0.63 in. from the outside surface; the next for a layer 1.53 in. from the outside surface; and the last for the inside surface. This chart illustrates clearly the decreasing amplitude and the lag of the temperature wave. The wave on the outside has the largest amplitude, occurring earliest, and the wave on the inside has the smallest amplitude, occurring latest.

The temperatures as shown were those obtained from the recording voltmeter. The curve for the inside surface was checked by measuring the current (representing the heat flow) from the inside surface to the air. These values, multiplied by the resistance representing the boundary (film conductance) equal the voltage difference between the *inside surface* and the *room air.* Hence, by adding to the product the (constant) value of the room-air temperature, the temperature of the inside surface may be obtained.

The curves in Fig. 6 give heat flow in relation to time. The time is plotted as abscissa, and the heat flow values as ordinates. The chart contains three curves: one for the heat flow through the outside surface as observed on the model; one for the heat flow from the inside surface (to the air in the room) as observed on the model; and a third for the heat flow from the inside surface as observed by direct thermal measurements at the ASHVE Research Laboratory.

The chart shows clearly the results of the mass effect of the wall. The crest of the heat flow as entering the outside surface is delayed and decreased in amount. The crest reaches the inside later and with a smaller amplitude.

The room temperature was held constant at 70.4 F. During the 24-hour period there was more heat flowing from the outside to the inside than vice versa. This *surplus* of *heat* was carried away by the cooling equipment which held the room temperature at 70.4 F. Without the cooling equipment in operation, the average temperature in the room would have been higher. Obviously, the heat flow (integrated over 24 hours) through the outside surface must equal the heat flow through the inside surface. Both values were measured independently; therefore a comparison of the two serves to check the accuracy of the method.

The total excess of heat flowing from the outside to the inside greater than

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the flow from the inside to the outside, as measured on the outside surface, was 123 Btu per square foot. Measurement on the inside surface showed a total heat of 117.5 Btu per square foot. The difference is 4.5 per cent.

Discussion of Results

1. The heat flow caused by the difference in temperature between the inside surface of the wall and the room-air (Fig. 6) shows greater deviations from the Pittsburgh

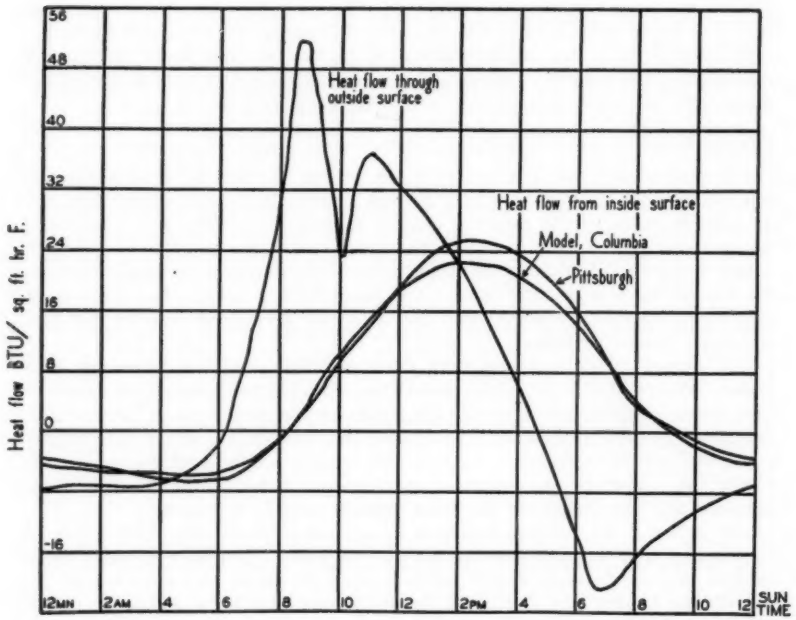

FIG. 6. HEAT FLOW THROUGH OUTSIDE SURFACE AND FROM INSIDE SURFACE; AIR TEMPERATURE IN ROOM HELD CONSTANT AT 70.4 F FOR EXPERIMENT A, 2-IN. PINE WALL

results than the temperatures. This is due to the fact that the heat flow is proportional to the temperature *difference*, which is found by deducting from the changing surface temperature the constant value of the air temperature.

2. The difference between the results obtained by the model method and those obtained at Pittsburgh are to be attributed to the following causes:

a. In Pittsburgh the room temperature was not constant but was subject to variations of plus or minus 2 per cent. This small variation may have a big influence on the heat flow for the same reasons as explained in paragraph 1.

b. The film conductance is *not* constant but changes with temperature. The higher surface temperatures probably cause a higher film conductance and consequently higher heat losses.

c. Manual adaptation of the voltage on the outside surface might have caused a curve slightly different from the value measured in Pittsburgh.

EXPERIMENT B

Problem

Given a building wall and the amount of solar radiation in relation to time for the outside surface, and the temperature of the air surrounding the outside surface of the wall. How does the heat flow from the inside surface change during one cycle if the inside air temperature is held at a constant value?

The experiment was conducted for a 13 in. brick wall for which the ASHVE Research Laboratory gave the following physical properties:

Thickness—13 in.
Thermal conductivity—5.2 Btu per square foot per degree Fahrenheit per hour, per inch.
Density—110.1 lb per cubic foot.
Specific heat—0.196 Btu per pound per degree Fahrenheit.
Film conductance (outside)—4.0 Btu per square foot per degree Fahrenheit per hour.
Film conductance (inside)—1.6 Btu per square foot per degree Fahrenheit per hour.

The thermal properties and film or air boundary conductance were assumed to be independent of temperature.

In order to get some indication as to the importance of the outside film conductance, a second set of experiments was run, with a boundary resistance representing a film conductance of only 2 Btu per square foot per degree Fahrenheit per hour.

The outside film conductance of 4 Btu per square foot per hour per degree Fahrenheit occurs at a wind velocity of 6 mph. The film conductance of 2 Btu per square foot per hour per degree Fahrenheit occurs at a wind velocity of 0.5 mph.

Procedure

The procedure was essentially the same as that used in Experiment A, the main differences being the following:

a. In order to represent the changing amount of heat input due to sun radiation, the current (milliammeter *M*, wiring diagram Fig. 7), was controlled.
b. The resistor $R_{B\text{-}o}$ was used to represent the film or boundary conductance at the outside wall.
c. One end of the resistor $R_{B\text{-}o}$ was connected to the end of the model representing the outside surface, the other to the power supply (*P.S.* II), supplying a changing voltage. This voltage was regulated according to a curve representing the outside air temperature.
d. As long as energy was being received from the sun, it was split into two parts: one flowing into the wall, and the other from the wall surface to the surroundings. After the radiation from the sun ceased, switch *Sw* was opened; and the wall was only in heat exchange with the surrounding air by means of the resistor $R_{B\text{-}o}$.

Results

The results are represented in a manner similar to those in Experiment A. The temperature time curves are shown in Fig. 8 with the time plotted as abscissa, and the temperatures as ordinates. Two groups of curve were drawn. One group, drawn with full lines, shows the temperatures for an outside film

conductance of 4 Btu per square foot per degree Fahrenheit per hour; the other, drawn with dotted lines, shows the temperatures for an outside film conductance of 2 Btu per square foot per degree Fahrenheit per hour. Each group consists of four curves: one for the outside surface; the next for a layer 4.33 in. from the outside surface; the next for a layer 8.67 in. from the outside surface; the last for the inside surface.

The lower portion of Fig. 9 shows the heat flow from the inside surface to the air in the room in relation to time. In the upper part of Fig. 9 the temperatures are plotted in relation to time. Time is plotted as abscissa for both sets of curves, while heat flow and temperatures are plotted as ordinates.

Three heat flow curves were drawn for the following conditions: one obtained on the model, for 2 Btu per square foot per degree Fahrenheit per hour

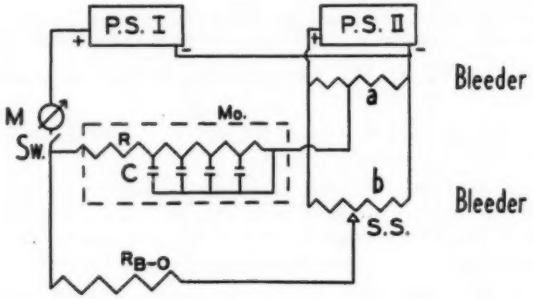

FIG. 7. CIRCUIT DIAGRAM FOR EXPERIMENT B

P. S. I=power supply I, supplying current, representing heat flow from the sun; heat flow is varied from 100 per cent to zero
P. S. II=power supply II feeding the two bleeders (potentiometer) resistors, supplying the auxiliary potentials.
a=voltage corresponding to the constant room air temperature of 78.8 F
S. S.=selector switch to adjust voltage *b*
b=voltage corresponding to the outside temperature
M=milliammeter
Sw=switch
Mo=model
R_{B-o}=resistance representing the film conductance from the outside surface to the surrounding air.

outside film conductance and constant room air temperature of 78.8 F; one for the same conditions as previous but for an outside film conductance of 4 Btu per square foot per degree Fahrenheit per hour; and one obtained in Pittsburgh on July 21, 1940, with an average outside film conductance of 4 Btu per square foot per degree Fahrenheit per hour, with the air temperature of the room being controlled by a thermostat, and oscillating as shown in the curves in the upper part of the chart.

Two temperature curves are shown in Fig. 9 with the temperature scale starting at 77 F. One indicates the inside surface temperatures as measured on the model, for an outside film conductance of 4 Btu per square foot per degree Fahrenheit per hour. This curve is an enlargement of the corresponding one shown in Fig. 8. The other curve shows the oscillations of the air-temperature in the room during the Pittsburgh tests on July 21, 1940.

Discussion of the Results

1. The amplitude of the temperature curve on the outside surface is flattened considerably by the thick wall so that the amplitude on the inside is much smaller. With an outside film conductance of 4 Btu per square foot per degree Fahrenheit per hour, the amplitude of $106.6 - 75.1 = 31.5$ F on the outside is flattened out to $81.9 - 80.1 = 1.8$ F on the inside, or 5.7 per cent of the outside surface amplitude. With an outside film conductance of 2 Btu per square foot per degree Fahrenheit per hour, the amplitude of $115.5 - 78.3 = 37.2$ F on the outside is flattened out to $82.5 - 80.8 = 1.7$ F on the inside, or 4.5 per cent of the outside surface amplitude.

2. The outside film conductance is of great influence on the heat flow from the inside surface, if the outside surface receives its heat as sun load. The total heat flow from the inside surface to the air in the room, which is kept at a constant value of 78.8 F by means of the cooling equipment is:

with a film conductance of 2 Btu per hour per square foot per degree Fahrenheit—
437.2 Btu
with a film conductance of 4 Btu per hour per square foot per degree Fahrenheit—
312.9 Btu

The ratio $\frac{437.2}{312.9} = 1.4$; that means an increase in outside film conductance from 2 to 4 (corresponding to an increase of wind velocity from 0.5 to 6 mph) reduces the cooling load to 60 per cent of its original value (at 2 Btu per hour per square foot per degree Fahrenheit). The reduction is not the same over the entire cycle of 24 hours. The discrepancy is largest at 1:15 P.M. (time of smallest heat flow: $\frac{3.02}{1.87} = 1.62$; the discrepancy is smallest at 10:45 P.M. (time of greatest heat flow) $= \frac{6.01}{4.61} = 1.30$. These experiments bring out clearly that the heat flow is subject to a much higher percentage variation over the 24 hours than the inside surface temperature.

3. The results of the model experiments can be compared in two aspects with the direct thermal measurements: a. The outside surface temperature, and b. The heat flow from the inside surface to the air in the room.

Outside Surface Temperature

Inasmuch as only the heat flow from the sun was given, the outside surface temperature results from the heat flow through the wall and to the outside air. The temperatures as measured in Pittsburgh and those found by the model method check closely. The largest discrepancy is 4 F, the average discrepancy is only 1.9 F.

Inside Heat Flow

The heat flow as observed in Pittsburgh does not check closely with the values measured by the model method. The minimum measured in Pittsburgh was 1.35 Btu per square foot per hour per degree Fahrenheit as against 1.87 as measured on the model. The values for the total amount of heat gain (heat flow integrated over a 24-hour period) differ only slightly. The *Pittsburgh*

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value is 290.98 Btu per 24 hours, while the *Columbia value* is 316.78 Btu per 24 hours.

The difference in shape of the two curves may be attributed to the difference of conditions prevailing during the two experiments. In Pittsburgh the room-air temperature was oscillating, while the model experiment was conducted with constant air temperature. Naturally the heat flow from the inside sur-

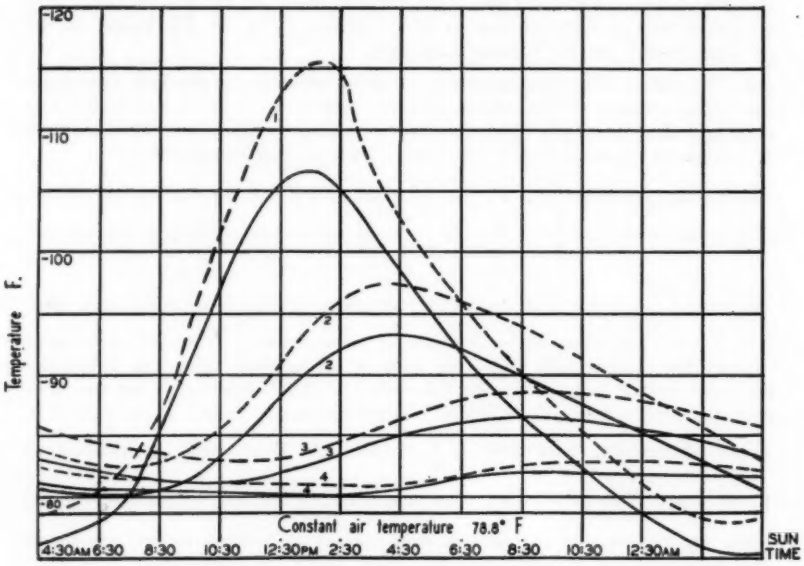

FIG. 8. TEMPERATURE-TIME CURVES; TEMPERATURE AT THE OUTSIDE AND INSIDE SURFACES OF THE WALL AND WITHIN THE WALL FOR EXPERIMENT B, 13-IN. BRICK WALL

Full lines for an outside film conductance of 4 Btu per square foot per hour per degree Fahrenheit. Dotted lines for an outside film conductance of 2 Btu per square foot per hour per degree Fahrenheit Lines:

1. at the outside surface
2. 4.33 in. from outside surface
3. 8.67 in. from outside surface
4. at the inside surface

face to the room-air is the result of the difference in temperature between the two values.

The shape of the temperature wave on the inside surface is almost independent of the small fluctuations of the room-air temperature and is caused entirely by the nature of the wall and the heat wave imposed on the outside surface. This temperature wave is in interaction with the wave of the air temperature. It is obvious that the resulting heat flow must differ from the flow obtained from the difference between the wave of inside surface temperature and a constant air temperature. This is still stressed by the fact that the temperature differences causing the heat flow are of the same order of magnitude as the oscillations in air temperature.

The differences in heat gain (heat flow integrated over a 24-hour period) can be explained by several factors. First, the inside film conductance assumed to be 1.6 Btu per square foot per hour per degree Fahrenheit might have actually been another value. The heat flow being directly proportional to the film conductance, would have had correspondingly different values. Furthermore, the measuring of small temperature differences (only 1 to 3 F) is

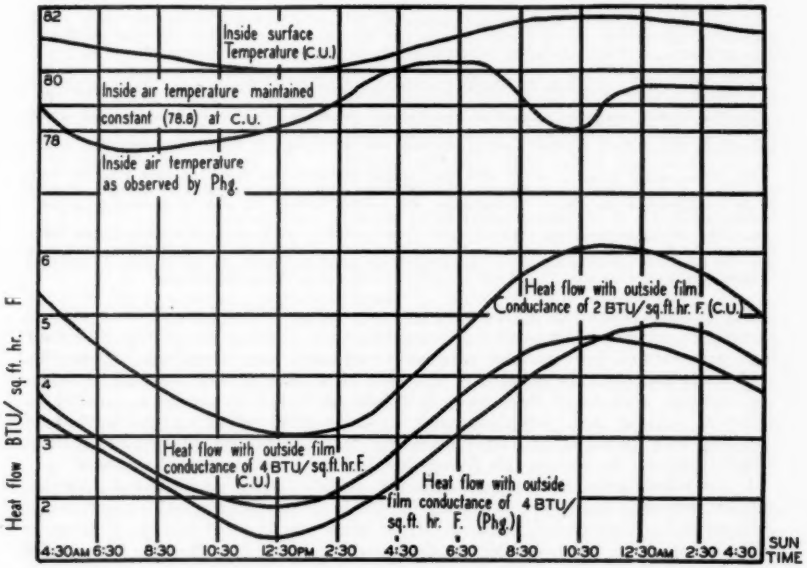
82
80
78
6
5
4
3
2
Inside surface Temperature (C.U.)
Inside air temperature maintained constant (78.8) at C.U.
Inside air temperature as observed by Phg.
Heat flow with outside film Conductance of 2 BTU/sq.ft.hr.F. (C.U.)
Heat flow with outside film conductance of 4 BTU/sq.ft.hr.F. (C.U.)
Heat flow with outside film conductance of 4 BTU/sq.ft. hr. F. (Phg.)
Heat flow BTU/ sq. ft. hr. F
4:30AM 6:30 8:30 10:30 12:30PM 2:30 4:30 6:30 8:30 10:30 12:30AM 2:30 4:30
SUN TIME

FIG. 9. RESULTS FROM EXPERIMENT B, 13-IN. BRICK WALL

Upper curves; inside air temperature
Lower curves; heat flow from inside surface to air in room

subject to larger proportional errors than the measurement of larger differences. It is believed that this error is greater in the case of the heat flow meter than in case of the model measurement.

CONCLUSIONS

The model method yields results which check closely with direct thermal measurements. Advantages of the model method are:

1. Elimination of building actual walls.
2. Considerably shorter time for experiments.
3. Elimination of the dependence on weather.
4. Possibility of maintaining constant air temperatures.
5. Possibility of investigating separately the influence of the different variables (*e.g.*, film conductance, etc.).

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DISCUSSION

C. A. ADAMS, Philadelphia, Pa. (WRITTEN): No engineering problem has been side-stepped more persistently than that of transient heat flow and no other problem has been crying out for solution more vigorously. The reasons for this side-stepping are:

1. The phenomena are very complicated from the theoretical point of view and involve mathematics beyond that available to the majority of engineers.
2. The constants involved are difficult to measure and are often anything but constant.
3. It is very difficult to visualize the phenomena in any quantitative sense, that is to get a quantitative perspective of the problem. Any one who has struggled with the mathematics of these problems and their application to specific cases will appreciate this statement.

There are three methods available for the solution of transient heat flow problems.

(a) The brutally empirical method of experimental measurement connected with a setup which duplicates with fair exactness the conditions surrounding the problem. This method is usually exceedingly difficult and expensive and serves only for the problem in hand. Moreover, the results are often questionable because of the difficulties involved.
(b) *Mathematical Solution*—Even with very simple problems this is not an easy job. In fact, for many of these problems a mathematical solution is impossible except with the aid of very crude simplifying assumptions. Nevertheless, this method is very valuable in developing a quantitative perspective of the relative importance of the major variables involved.
(c) *The Model Method*—Of the two types now available, hydraulic and electrical, the latter developed by Dr. Paschkis is for many reasons superior. It should be noted, however, that even here the degree of complication in the geometry of the problems which can successfully be attacked is limited, although not nearly so much so as is true of the mathematical method.

The problem discussed in the subject paper being of the parallel flow type is exceedingly simple as compared with very important problems in welding with their high temperatures, more variable parameters and much more complicated geometries.

Nevertheless, since the solution of this type of problem is often so vital to industry, research work in all three methods should be pushed as rapidly as possible by figuring, checking and rechecking until we have developed a rational method of handling such problems and have developed that quantitative perspective, without which there can be nothing but floundering and expensive industrial mistakes.

It has often happened in other fields that in spite of the mathematical difficulty, rational formulae have been developed in which the coefficients and exponents are determined by experimental research. If in doing this, careful account is taken of the relation of the minor variables to these coefficients, results can often be extended well beyond the range of the experimentation. This is the ideal towards which we should work in this very important field of engineering endeavor.

Finally, I consider Dr. Paschkis' work a very important part of this picture and hope that he will be given all possible encouragement and financial support.

J. R. ANDERSEN, Philadelphia, Pa. (WRITTEN): The two problems of periodic heat flow in building walls discussed in this paper were presumably chosen to indicate the validity of this electric analogy method in the solution of periodic heat flow problems. The author has shown that it is possible to represent electrically all of the essential features of the thermal performance of building walls by a simple electric model circuit.

There are in general three methods which may be applied to the solution of engineering problems (*a*) analytical methods, (*b*) direct experiments on the equipment, and (*c*) model experiments. All three of these methods have their own sphere of usefulness, but each when correctly applied is capable of producing essentially the same information.

Ordinarily the analytical solution of periodic heat flow in walls and slabs is practical only when (*a*) the wall is homogeneous and isotropic, (*b*) when the surface boundary conditions are readily expressible mathematically. The electric analogy discussed in this paper is not subject to these limitations since non-homogeneity and non-isotropy can be built into the model and most boundary conditions can be simulated by manipulation of the model controls.

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The two problems considered in this paper are susceptible of analytical solution but the computations would be rather tedious.

The theoretical basis for the electrical model discussed is, as the author points out, the identity of form of the heat flow equation and that of the non-inductive electric circuit. The author neglects to mention, however, that the models described do not strictly conform to that of the non-inductive transmission line as required by the strict application of the electrical analogy, but the model has been simplified by a lumping process for the capacitors. Strictly speaking the models described represent a series of slabs of infinite thermal conductivity but finite heat capacity separated by boundary resistances with no heat capacity. Such a system represents, in the limit as the number of such slabs is increased without limit, the actual thermal problem under consideration. It is fortunate indeed that with the simple models used the performance of the walls under consideration can be so nearly duplicated.

In conclusion the author is to be congratulated on the introduction of this interesting analogy to the ASHVE and further results from this method will be looked for with considerable interest.

A. L. LONDON, Stanford University, Calif. (WRITTEN): The method described by Dr. Paschkis is indeed a very powerful experimental technique for solving thermal conduction transient problems. However, it should also be noted that the behavior of a simple lumped parameter capacity-resistance circuit may be determined analytically. Thus the performance of an idealized lumped parameter circuit consisting of three or fewer capacity lumps can usually be predicted without recourse to experiment. For those thermal systems requiring more than three capacitor lumps for an adequate idealization, analytical solutions are still obtainable by the use of a differential analyzer.[6]

In application of the experimental method the readily obtainable analytical solution of a simple capacity-resistance circuit may be employed to provide a valuable verification of the suitability of the model instrumentation, circuit arrangement, and capacitor characteristics (leakage). For instance, the circuit of Fig. 4 in the paper may be rearranged, employing the same elements, to a single capacity system representing the first approximation of the thermal prototype.

For regular periodic excitation the behavior of this simple system can be readily obtained analytically. A simple sinusoidal excitation, of about the same amplitude and period as to be employed in the model tests, may be impressed on the single capacitor system and the behavior measured by the instrumentation. If the measured behavior compares favorably with that predicted analytically, it may be concluded with reasonable certainty that the four capacitor model, together with the instrumentation, will yield satisfactory results in so far as the idealized model represents the prototype.

For many problems a first or second approximation lumped parameter thermal circuit can be devised, the response of which is conservative from the point of view of the problem in mind. The analytical solution of the simple circuit is in many cases of sufficient accuracy for design purposes. For instance, the problem of five partitions of structural members can be treated as indicated. It is also feasible for some systems to provide two idealized circuits which will in behavior straddle the response of the actual system. Then if the two idealized behaviors are in close agreement it can be concluded that either one is an adequate approximation; and, of considerable importance, some idea of the accuracy of the predicted behavior is thereby obtained. A similar procedure could be employed with the electrical model technique.

L. M. K. BOELTER,[7] Berkeley, Calif. (WRITTEN): The solution of problems by analogy is a powerful weapon. The differential equations of two systems, as well as the boundary and initial conditions, must be similar in order that the two systems be analogous. Thermal and electrical circuits can be designed which are strictly

[6] V. Bush, J.F.I., Vol. 212, No. 4, pp. 447-488, October, 1931.
[7] Professor of Mechanical Engineering, University of California.

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analogous. The analogous variables and system parameters noted in paragraph four of the paper do not represent the only analogous systems; for instance, an electrical circuit consisting of resistance and inductance can be arranged to satisfy all of the conditions for analogy. Incidentally, the word *transient* seems to be gratuitous as used in paragraph 4.

It should be emphasized that the lumped parameter electrical circuit shown in Fig. 1 is only an approximation to the thermal system of distributed parameters shown in the same figure. Electrical circuit analysts have made available many *equivalent* lumped parameter circuits. Finally the contact resistances of the thermal system are not represented in the electrical circuit.

A discussion of the thermal balance under *Procedure* seems to overlook the energy reflected from the walls during the day (which indeed may be small) as well as that radiated from the walls. The last sentence, "After the radiation from the sun ceased, switch *Sw* was opened; and the wall was only in heat exchange with the surrounding air by means of the resistor R_{B-o}" is almost correct, but omits consideration of the fact that the transfer from the wall is controlled by convection and irradiation of the wall by space, nor does this sentence include mention of the radiosity of the surface.

Dimensionless ratios similar to those used to express the periodic (or transient) heat flow through a thermal system may also be written for the analogous electrical system. The latter ratios allow the rapid calculation of the parameters of the analogous system. Fourier's Modulus $\frac{a\Theta}{L^2}$, for instance, becomes Θ/crL^2, where:

a = thermal diffusivity
Θ = time
L = length of thermal system
= length of electrical circuit
c = electrical capacity per unit length
r = electrical resistance per unit length

The heat flow into a thermal system as measured by a heat meter will differ from the actual heat rate, due to the thermal capacity and resistance of the meter, the contact resistance between the wall and the meter, the difference between the wall and meter emissivities, as well as the distortion of the thermal flow path through the wall. These corrections can readily be computed for a given system-meter application, and will no doubt account for some of the discrepancies.

H. B. NOTTAGE, East Hartford, Conn.: I have two questions regarding the definition of the experimental system which was used here for the analogy. What is the basis for selecting the number of capacitor units that are employed to approximate the actually distributed capacity of the thermal system, and how does this selection affect the accuracy of the measures?

Second, in the electrical model experiment, it appears that the voltage enclosed would vary in certain finite steps, whereas, in the actual thermal system, the temperature potential is some continuous function of time. How does this affect the accuracy of the measurements?

F. C. HOUGHTEN, Pittsburgh, Pa.: This fortunate development will greatly decrease the work required in establishing the curves which you will use for estimating heat gains. The Research Laboratory paper (Heat Gain Through Walls and Roofs as Affected by Solar Radiation) to be presented next at this session will show just how laborious that process is, and how long it would take to supply all the information needed for THE GUIDE. Dr. Paschkis was able in a few minutes to determine results that took the Laboratory at Pittsburgh months to accomplish by actual observation.

In a previous paper [8] by the Laboratory a few years ago there was presented a

[8] Loc. Cit. See Note 1.

series of mathematical analyses which would satisfactorily do this job, but it took fifteen or twenty hours on the part of a pretty good mathematician to develop a single one of these curves. It was necessary in that study, as in any mathematical analysis, to feed constants into the analysis, and as a result an answer in the form of a heat gain curve was had. The answer was no better than the constants; the same is true here.

Before Dr. Paschkis can use his apparatus to determine the curve for any given wall, he must have constants for all the different materials that go into the wall. The one or two walls already used were fairly homogeneous, and it remains to be seen how well this same application will work out in more complicated walls, including walls with air spaces. There is no doubt whatever that it will work, but it will probably take a little more cooperation between a laboratory which will determine these curves experimentally and Dr. Paschkis with this electrical analytical method, in order to determine just how some of those constants are to be evaluated and used. Surely after a little work of this kind it should be possible to determinine the heat flow curve for any wall listed in the ASHVE GUIDE, no matter how complicated in construction, when the conductivity and heat capacity of the materials in it and the temperature cycle imposed on one side are known.

Another immediate use to which this analytical method may be put is the determination of the effect of certain combinations and arrangements of such factors as resistance to heat flow, or insulating value, and thermal capacity. Laboratory work indicates that the relationship between these two factors in their effect on the heat flow curve is not simple. The development of a complete understanding of these relationships by laboratory experimentation would be almost interminable. A few simple analyses by Dr. Paschkis with well chosen combinations of the two factors should clear this matter up. Again, the relative effect of placing insulation on the inside or outside of a relatively high heat capacity wall is not well understood. This method unlike laboratory experimentation should make it possible with relatively little work to obtain a conclusive answer. There are many other similar general questions of cyclic heat flow which may be worked out, pending the establishment of a satisfactory background in the method and values to be used in establishing usable curves, for actual wall combinations not already determined by laboratory experimentation.

PERCY NICHOLLS, Pittsburgh, Pa.: I think that the previous discussers have shown that the method described is correct provided the proper constants have been evaluated. However, Dr. Paschkis' method would have been of little value unless these constants had previously been obtained by experimental work, or by direct measurement. I think he has developed a tool which will have a great number of useful applications. I mentioned this many years ago in connection with high temperature furnace work, where I suggested that in connection with determination of heat loss through a furnace structure, the mathematics were so complicated that it would be impossible to solve each case. At that time we calculated a number of solutions for typical sets of conditions and then anyone desiring to estimate the heat loss from a given furnace structure would be able to determine it by one of these solutions.

With Dr. Paschkis' method a more exact result can be obtained and I believe the Society should cooperate with him to the extent of outlining some investigations which will be of great value to engineers in our profession.

DR. PASCHKIS: In the discussions, two issues are raised repeatedly: (1) the fact that we have applied lumped circuits and the number of lumps (sections); and (2) the influence of correct selection of the constants which are fed in the electric analogue.

Lumping. The fact that the method is using lumped circuits rather than one with evenly distributed resistance and capacitance has been dealt with in a previous publication, quoted in this paper.[9] It is obvious that the accuracy of the method is improved with an increase in the number of sections. Fortunately results of reasonable accuracy are obtained with a relatively small number of sections. In one case

[9] Loc. Cit. See Note 2.

(of two-dimensional flow), we operated with four sections in one direction and three in the other and obtained results within three per cent of the values found by mathematical analysis. For the experiments dealt with in the present paper, we have used 8 sections. Experiments have been started to determine the necessary number of sections in a more general way.

Constants. The assertion by Dr. Houghten is undoubtedly correct, that if the assumption of the physical constants is incorrect, the results of the experiments will also be incorrect. Because of the uncertainty about these constants, best results will be achieved by combining direct thermal measurements with investigations by the model method. One point should, however, be mentioned in this connection. The rapidity with which experiments are carried out by the electrical method allow the repetition of a test with various values of the constants. In general three such constants are involved: thermal conductivity, specific heat, and boundary conductance. By systematically varying these values in consecutive experiments, the relative importance of each can be determined. Efforts toward more exact knowledge of the constants can be concentrated on the one bearing most on the results.

Stepwise change of voltage (Mr. Nottage): If the steps are taken sufficiently close, the accuracy will be sufficient for practical purposes. For extremely rapid changes, more steps are necessary, and therefore the time scale in the model experiment may have to be changed, in order to allow sufficient time for many steps.

Inductance (Professor Boelter): The general conception is that while electrical capacitance and resistance have their strict analogies in the realm of heat, that is not true for inductance. It would be interesting to learn from Professor Boelter which heat phenomenon he considers analogous to inductance. Film conductance: The resistor R_{B-0} comprises the surface losses by convection as well as by radiation. If necessary the resistor can be made variable in order to represent changes in the over-all film conductance. Dimensionless ratio: Reference is here made to the previous publication, mentioned in the present paper.[10] Heat flow meter: Theoretically the observations of Professor Boelter are correct. The practical design of the heat flow meter is, however, such that the resulting change in apparent thermal capacity and resistivity of the wall are negligible. This is evident from the various previous investigations by Dr. Houghten and co-workers, comparing direct thermal measurements with mathematical analysis.

Accuracy (Professor London): These remarks are of great theoretical interest. In view of repeated experiments, checking the electric method with thermal measurements and with mathematical results, further proof however does not seem necessary. In most cases, the accuracy of one or two lumps will not be sufficient, thus ruling out mathematical analysis. But even where this accuracy is sufficient, the electrical method will yield more rapid results.

[10] Loc. Cit. See Note 2.

No. 1195

HEAT GAIN THROUGH WALLS AND ROOFS AS AFFECTED BY SOLAR RADIATION

By F. C. HOUGHTEN,* E. C. HACH,** S. I. TAIMUTY,*** and CARL GUTBERLET,†
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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in its Research Laboratory at the Pittsburgh Experiment Station of the U. S. Bureau of Mines.

HEAT gain through walls to summer cooled and air conditioned space is complicated by the wide cyclic variation in the outside air temperature and sun intensity. This is very marked when compared with the relatively constant flow through the same walls from a winter heated and air conditioned space.

When applied to winter heating the design outside temperature is likely to be as low as zero with a daily fluctuation of approximately 10 deg, whereas that maintained indoors would be 70 F or higher, giving a daily variation of about one-seventh of the temperature difference motivating heat flow through the wall. In summer cooling the design outside air temperature will vary with the locality over an approximate range of from 90 to 105 F with 95 as a good average for most metropolitan areas. With an inside maintained dry-bulb temperature of 75 F or higher, this gives a motivating temperature difference of 20 deg or less which usually fluctuates by more than this difference; that is, the outside air temperature at night usually falls below that maintained indoors. Solar radiation is a powerful factor in further affecting a wide cyclic variation in the heat flow through a wall during the summer, while in winter it has a relatively smaller effect and then tends to diminish rather than increase the magnitude of the heat flow. Hence, while it is relatively satisfactory to use simple transmission coefficients or U values applying to steady state flow for winter heating, such factors and assumptions cannot be used with safety for summer cooling calculations.

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Several papers [1, 2, 3] resulting from work carried on at the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS have served to bring out this fact. By 1939 these studies had made it sufficiently obvious that the generally accepted winter heat flow coefficients could not be applied to summer cooling, to bring about the organization of a study planned by the Technical Advisory Committee on Cooling Load in Summer Air Conditioning [4] and aimed to make available the necessary data for calculating such gains. An 18 ft x 18 ft x 8 ft high cubicle,[5] was built, in which the simultaneous rates of heat gain through three different walls, having three or four orientations, and nine horizontal roof decks could be studied. A brief report [6] covering certain phases of the study, dealing with sprinkled and flooded roofs was presented in 1940 and the results of further studies in this cubicle are given in this report.

DESIGN DATA FOR WALLS AND ROOFS STUDIED

The original cubicle contained 5½ ft wide x 7 ft high wall sections of 12 in. brick and plaster and 4 in. brick, 8 in. hollow tile and plaster, facing north, east, south and west, and a third section of 4 in. brick and frame veneer with all but the northern exposure, gave a very satisfactory direct comparison of the time—heat flow relationships for the same wall on different exposures. During the summer of 1941 the work was continued with some of the original walls replaced by others, but not giving the direct comparison for different orientations. It is believed, however, that the facts learned in connection with the original walls make it possible to establish the time—heat flow relationships for different exposures when that for one exposure is known. Tables 1 and 2 give the construction, orientation, resistance to heat flow and calculated heat capacity for all walls and roofs studied. Figs. 1 to 6 give heat flow—time relationship curves for each of the side walls studied for four orientations and for each of the horizontal roof panels, all for design weather conditions of temperature and solar radiation on August 1; that is, for an August 1 day when the outside temperature reaches a high of 95 F dry-bulb and the solar radiation is of design intensity.

Design Air Temperature and Solar Radiation

The design outside dry-bulb temperatures to be applied to the estimated cooling load in summer air conditioning have been arrived at for different metropolitan districts by committees of the Society and published in THE GUIDE 1942. These values are the maximum temperatures for days occurring with sufficient frequency to be used for design purposes. In this study as in

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[1] Heat Transfer through Roofs under Summer Conditions, by F. C. Houghten and C. G. F. Zobel. (ASHVE TRANSACTIONS, Vol. 34, 1928.)

[2] Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh, and Paul McDermott. (ASHVE TRANSACTIONS, Vol. 38, 1932.)

[3] The Effect of Solar Radiation on the Heat Transmission Through Walls, by F. C. Houghten, Carl Gutberlet, and Albert A. Rosenberg. Symposium on Thermal Insulating Materials, March 8, 1939. (*American Society for Testing Materials*, Philadelphia, Pa.)

[4] C. M. Ashley, *Chairman*; John Everetts, Jr., E. H. Hyde, C. S. Leopold, C. O. Mackey, R. M. Stikeleather, J. H. Walker, W. E. Zieber.

[5] Summer Cooling Load as Affected by Heat Gain Through Dry, Sprinkled and Water Covered Roofs, by F. C. Houghten, H. T. Olson and Carl Gutberlet. (ASHVE TRANSACTIONS, Vol. 46, 1940.)

[6] Loc. Cit. See Note 4.

TABLE 1—CONSTRUCTION, RESISTANCE TO HEAT FLOW, AND HEAT CAPACITY OF WALLS AND ORIENTATION IN WHICH THEY ARE TESTED

WALL NO.	CONSTRUCTION	ORIENTATION		*U* BTU PER SQ FT PER HOUR PER DEG F AIR TEMP. DIFF.	HEAT CAPACITY BTU PER SQ FT PER DEG F
		1940	1941		
1	8 in.[a] Brick, Plaster[b]	...	E	0.46	15.87
2	12 in.[a] Brick, Plaster	N, E, S, W	N, E	0.34	23.38
3	12 in. Brick, Furring Space,[c] 1 in. Structural Insulation Board, Plaster	...	S	0.14	23.78
4	16 in.[a] Brick, Plaster	...	E	0.27	31.80
5	8 in. Concrete[e]	...	W	0.69	23.22
6	4 in. Black, 8 in. Tile, Plaster	N, E, S, W	N	0.33	16.37
7	4 in. Brick, 8 in. Tile, Furring Space, 1 in. Structural Insulation Board, Plaster.	...	S	0.14	16.77
8	4 in. Brick, Wood Sheathing,[c] Stud Space,[d] Metal Lath, Plaster	E, S, W	...	0.28	8.31
9	4 in. Brick, Wood Sheathing, Stud Space, 1 in. Structural Insulation Board, Plaster	...	S	0.15	8.71
10	Wood Siding[f] Paper, Wood Sheathing, Stud Space, Metal Lath, Plaster	...	W	0.26	2.79
11	Wood Siding, Paper, Wood Sheathing, Stud Space Filled with Rock Wool Insulation, Metal Lath, Plaster	...	W	0.072	3.75

[a] Thickness of brick not including mortar, outside surface red mingled textured face brick.
[b] Plaster applied to brick, tile, and structural insulation board, consisted of ground coat and white skin coat $\frac{1}{16}$ in. thick; when applied to metal lath ground coat and skin coat, $\frac{3}{4}$ in. thick.
[c] Actual thickness, $\frac{33}{32}$ in.
[d] Actual thickness, $3\frac{5}{8}$ in.
[e] Poured concrete with rubbed surfaces, inside and outside.
[f] $\frac{3}{4}$ in. × 8 in. lap siding, painted white.

others recently made, it was desirable not only to establish the maximum temperature but also the typical time-temperature curve for the 24-hour period. The result of an analysis of the weather bureau reported hourly temperatures for a number of days when the maximum reached 95 F established such a curve given in an earlier Laboratory report.[7]

[7] Heat Gain Through Glass Blocks by Solar Radiation and Transmittance, by F. C. Houghten, David Shore, H. T. Olson and Burt Gunst. (ASHVE TRANSACTIONS, Vol. 46, 1940.)

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TABLE 2—CONSTRUCTION, RESISTANCE TO HEAT FLOW AND HEAT CAPACITIES OF ROOFS

ROOF NO.	CONSTRUCTION[a]	*U* BTU PER SQ FT PER HOUR PER DEG F AIR TEMP. DIFF.	HEAT CAPACITY BTU PER SQ FT PER DEG F
1	2 in. Concrete, Felt, Pitch, Smooth Black Finish[b]	0.40	6.92
2	No. 1 Insulated[c]	0.17	7.32
3	2 in. Concrete, Felt, Pitch and Slag	0.31	7.41
4	No. 3 Insulated	0.15	7.81
5	2 in. Concrete, Felt, Double Pitch and Slag[d]	0.26	7.92
6	2 in. Concrete, Felt, Pitch and Gravel Insulated	0.15	7.72
7	6 in. Concrete, Felt, Pitch and Slag	0.28	19.34
8	No. 7 Insulated	0.15	19.74
9	2 in. Reinforced Gypsum, Felt, Pitch and Slag	0.24	4.42
10	No. 9 Insulated	0.13	4.82
11	4 in. Reinforced Gypsum, Felt, Pitch and Slag	0.18	6.89
12	No. 11 Insulated	0.11	7.29
13	4 in. Tile, Felt, Pitch and Slag	0.25	5.80
14	No. 13 Insulated	0.14	6.20
15	2 in. Plank, Felt, Pitch and Slag[e]	0.20	3.22
16	No. 15 Insulated	0.12	3.62
17	2 in. Plank, Felt, Double Pitch and Slag	0.18	3.19
18	2 in. Plank, Felt, Pitch Smooth Black Finish Insulated	0.11	2.93

[a] For more complete description see earlier laboratory report.[f]
[b] Water proofing on all roofs consists of 4 ply felt set in pitch except plank roofs which consists of 5 ply.
[c] Insulated roofs had 1 in. structural insulation board under water proofing.
[d] Double pitch and slag roofs had 1st layer of slag filled with pitch and then 2nd layer or slag.
[e] 1 ½ in. matched yellow pine plank.
[f] Loc. Cit. See Note 4.

The Laboratory has found that it is desirable to arrive at design solar radiation curves for different vertical walls facing east, south, and west and for horizontal surfaces. Based upon pyrheliometer observations at the Laboratory and additional observations made over a period of years by the Pittsburgh Station of the U. S. Weather Bureau, solar intensities which might be expected to occur on at least a few days of each summer were arrived at. This includes not only the direct solar radiation intensity received in straight lines from the sun but also scattered or sky radiation. These design solar radiation

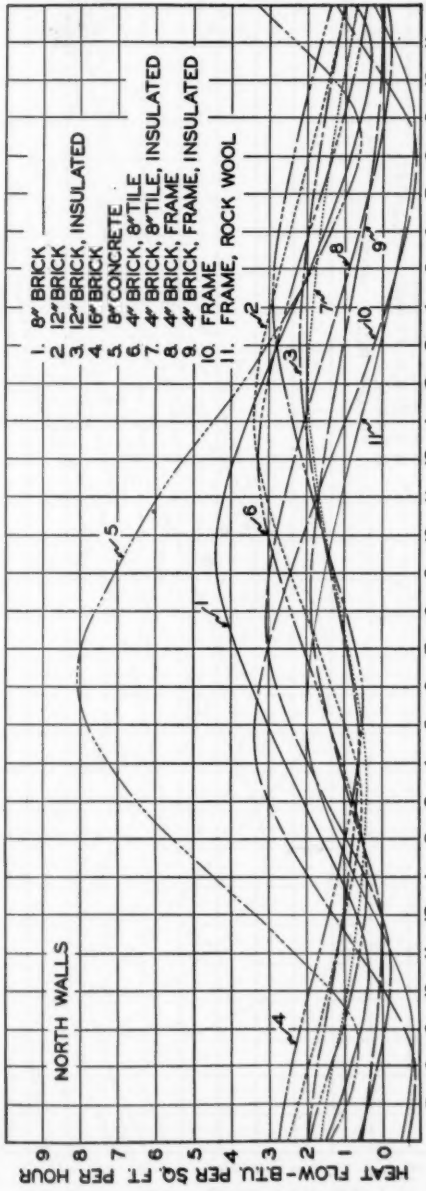

FIG. 1. HEAT FLOW-TIME RELATIONSHIP FOR NORTHERN EXPOSED WALLS

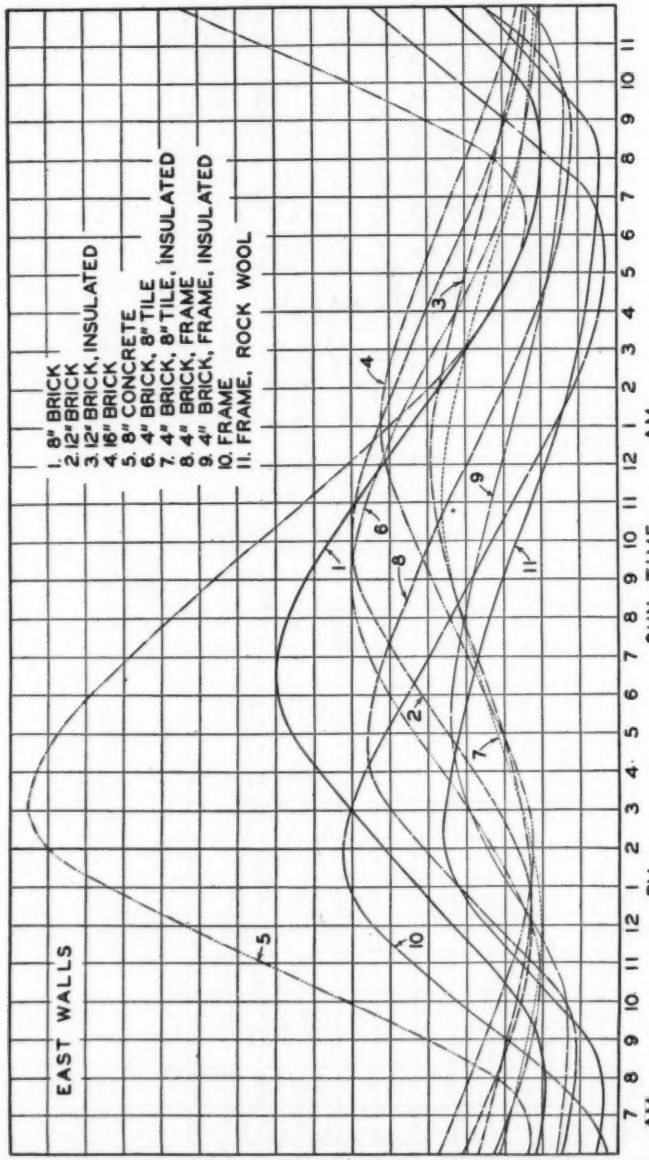

FIG. 2. HEAT FLOW-TIME RELATIONSHIP FOR EASTERN EXPOSED WALLS

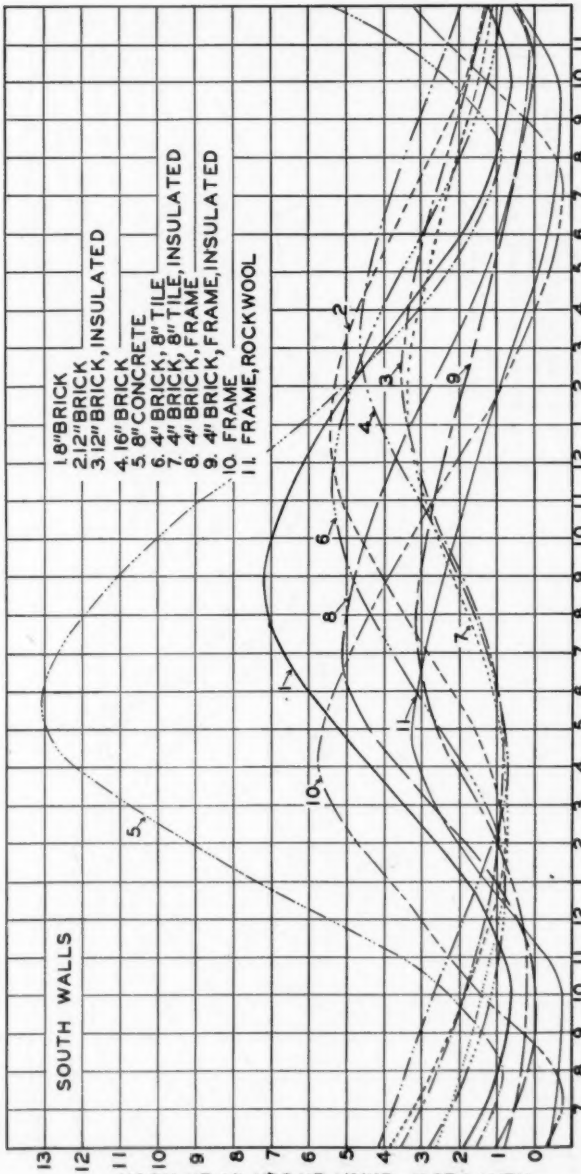

FIG. 3. HEAT FLOW-TIME RELATIONSHIP FOR SOUTHERN EXPOSED WALLS

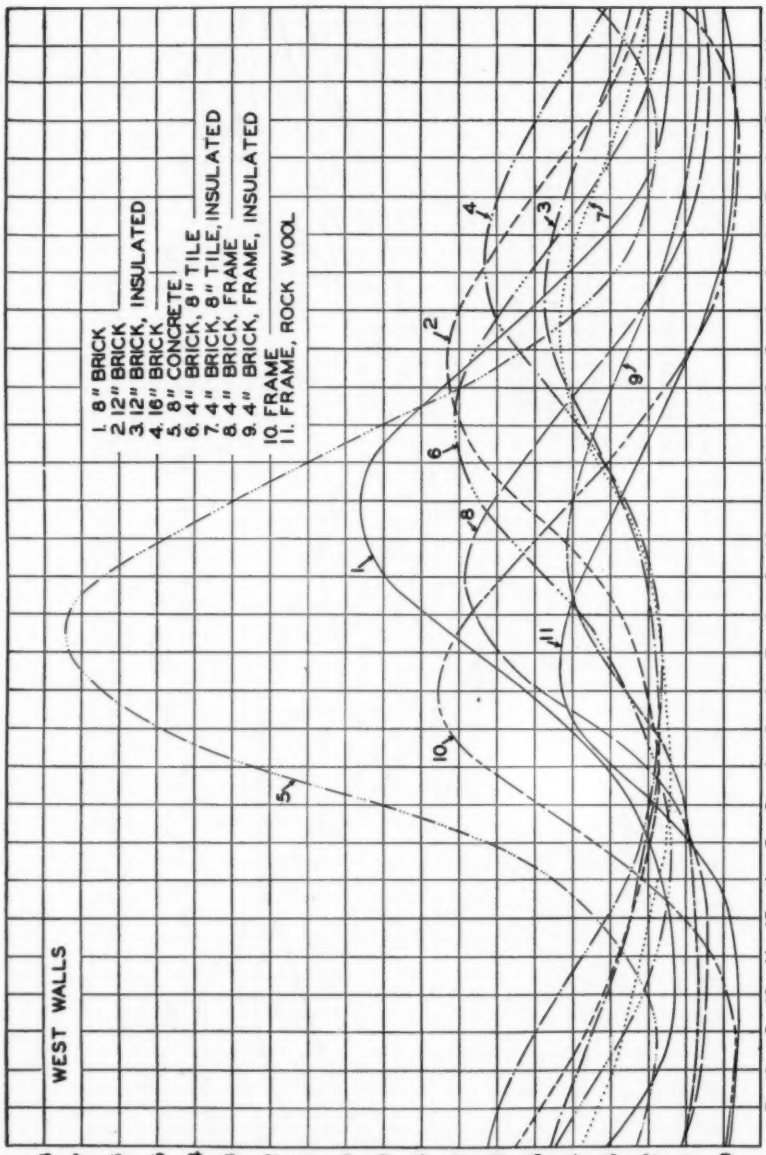

FIG. 4. HEAT FLOW-TIME RELATIONSHIP FOR WESTERN EXPOSED WALLS

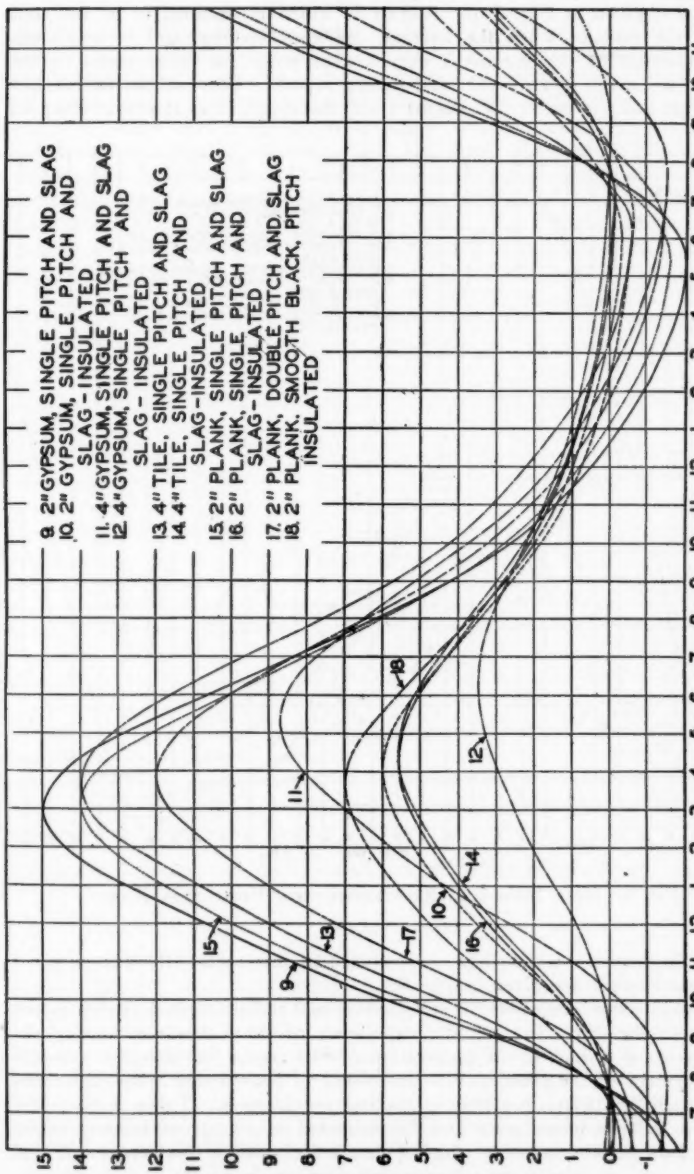
HEAT FLOW-BTU PER SQ. FT. PER HOUR
SUN TIME
P.M.
A.M.
9. 2" GYPSUM, SINGLE PITCH AND SLAG
10. 2" GYPSUM, SINGLE PITCH AND SLAG-INSULATED
11. 4" GYPSUM, SINGLE PITCH AND SLAG
12. 4" GYPSUM, SINGLE PITCH AND SLAG-INSULATED
13. 4" TILE, SINGLE PITCH AND SLAG
14. 4" TILE, SINGLE PITCH AND SLAG-INSULATED
15. 2" PLANK, SINGLE PITCH AND SLAG
16. 2" PLANK, SINGLE PITCH AND SLAG-INSULATED
17. 2" PLANK, DOUBLE PITCH AND SLAG
18. 2" PLANK, SMOOTH BLACK, PITCH INSULATED

FIG. 5. HEAT FLOW-TIME RELATIONSHIP FOR HORIZONTAL ROOFS

curves are given in Fig. 7 for August 1, which is assumed to be the time during the summer when the average combined cooling load is maximum. A comparison of the August 1 curves with those for other times of the summer are given in an earlier Laboratory report.[8] For comparison between the design solar intensity for August 1 and the actual solar radiation observed

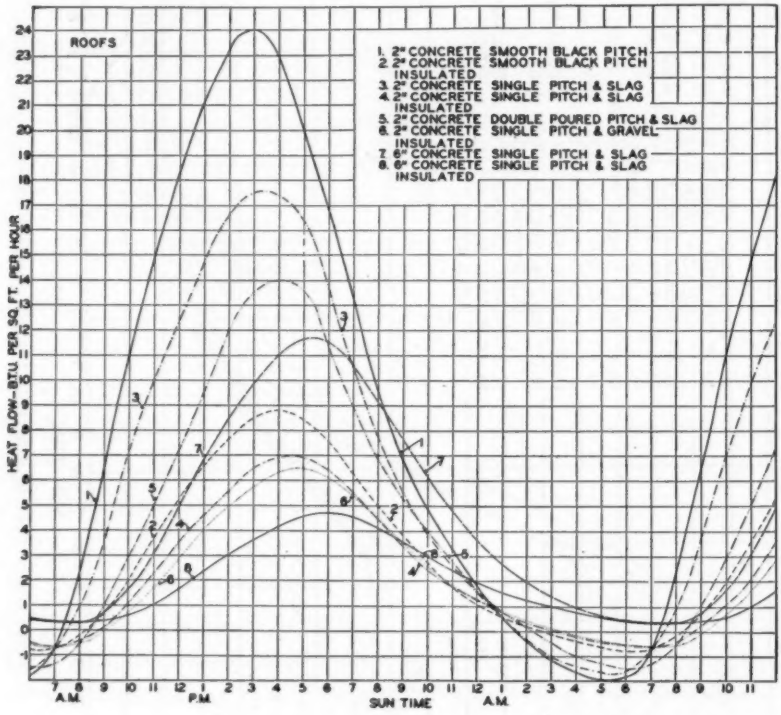
ROOFS
1. 2" CONCRETE SMOOTH BLACK PITCH
2. 2" CONCRETE SMOOTH BLACK PITCH INSULATED
3. 2" CONCRETE SINGLE PITCH & SLAG
4. 2" CONCRETE SINGLE PITCH & SLAG INSULATED
5. 2" CONCRETE DOUBLE POURED PITCH & SLAG
6. 2" CONCRETE SINGLE PITCH & GRAVEL INSULATED
7. 6" CONCRETE SINGLE PITCH & SLAG
8. 6" CONCRETE SINGLE PITCH & SLAG INSULATED
HEAT FLOW-BTU. PER SQ. FT. PER HOUR
A.M.
P.M.
SUN TIME
A.M.

FIG. 6. HEAT FLOW-TIME RELATIONSHIP FOR HORIZONTAL ROOFS

on certain days for which data are included in this report, the curves for a horizontal surface are given in Fig. 8.

Few days occur when both air temperature and radiation reach design values throughout the day. In fact, the experience of the Laboratory during the past 2 years as gained in the preparation of this report indicates that possibly the acceptance of the simultaneous occurrence of both design temperature and design radiation gives too high a cooling requirement. Table 3 gives the maximum outside temperature and the estimated percentage of solar radiation observed on certain days for which data are included in this report. It will

[8] Loc. Cit. See Note 7.

readily be seen that the coincidental occurrence of design conditions for both temperature and radiation when based solely upon chance will rarely occur, and further there is some indication and some reason to believe that the lack of concurrence is more than a matter of pure chance for it is possible that extremely warm days are generally associated with the occurrence of cumulous clouds during a part of the day. Haze and clouds observed in Pittsburgh on hot days during this study will probably occur in most industrial metropolitan areas and reduce the solar radiation intensity. There will be exceptions, however, where haze and clouds are less prevalent. This matter should be further studied with the possible reduction of the design heat gain curves here reported, as well as those for localities with other climates and latitudes.

In arriving at the design heat flow curves, Figs 1 to 6, inclusive, curves for several days when both the outside temperature and solar radiation were near design were studied and adjusted by extrapolation. For those walls for which

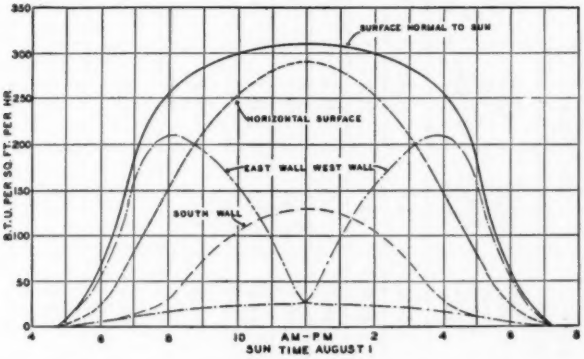

FIG. 7. DESIGN SOLAR RADIATION ON DIFFERENT SURFACES

data were obtained for only one orientation the curves were drawn from the relationships found from the few walls studied with more than one orientation.

TEST OPERATION

Each of the 11 side walls and 9 roof panels were equipped with a 2-ft square Nicholls' heat flow meter on the center of the inside surface. At frequent intervals throughout the 24-hour day, for extended periods including all types of summer weather, the following observations were made: heat flows; the outside air temperature 6 in. from each wall and 6 in. above each roof, shielded so as to minimize the effect of solar radiation; the outside surface temperatures; the inside surface temperatures; the air temperature at the control point in the room; and a number of temperatures within the individual walls and roofs. A shielded aspirating psycrometer was used to observe the outside air wet- and dry-bulb temperatures unaffected by the building. Fine wire copper-constantan thermocouples and a precision potentiometer were

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TABLE 3—WEATHER AND HEAT GAIN DATA DURING A HOT WAVE

DATE	MAXIMUM TEMP.	PER CENT[a] SOLAR RADIATION	WEATHER	UNINSULATED EAST WALL		INSULATED SOUTH WALL		WEST WALL			
				12 IN. BRICK[b]		12 IN. BRICK[c]		FRAME[d]		FRAME, ROCK WOOL[e]	
				MAX. HEAT FLOW BTU/SQ FT PER HOUR	TOTAL HEAT GAIN BTU/SQ FT PER DAY	MAX. HEAT FLOW BTU/SQ FT PER HOUR	TOTAL HEAT GAIN BTU/SQ FT PER DAY	MAX. HEAT FLOW BTU/SQ FT PER HOUR	TOTAL HEAT GAIN BTU/SQ FT PER DAY	MAX. HEAT FLOW BTU/SQ FT PER HOUR	TOTAL HEAT GAIN BTU/SQ FT PER DAY
7–23–41	88.5	75	Clear	3.0	32.2	1.0	13.1	4.4	18.2	2.5	13.8
7–24–41	91.0	50	Clear	3.8	40.9	2.0	18.9	5.9	39.4	4.3	31.2
7–25–41	95.0	51	Clear	3.9	48.4	1.9	27.2	5.8	44.0	4.4	34.6
7–26–41	97.0	50	Clear	3.9	66.4	1.8	33.8	4.8	38.4	3.4	32.4
7–27–41	100.2	52	Clear	4.5	64.2	2.2	38.2	4.6	39.4	2.4	28.0
7–28–41	96.0	60	Clear to Cloudy	4.4	82.0	1.9	39.8	5.0	48.4	3.6	36.4
7–29–41	92.2	40	Clear to Rain	4.9	89.8	2.3	43.0	2.9	27.8	2.0	24.4
7–30–41	94.0	86	Clear to Cloudy	5.1	79.0	1.8	30.2	5.4	31.6	3.8	23.8
7–31–41	87.0	40	Cloudy to Rain	No Heat Flows observed							

[a] Per cent of the integrated total for the day.
[b], [c], [d], [e] Walls No. 2, 3, 10 and 11, Table 1.

used for observing all other temperatures. A recording potentiometer provided a record of two, and in some cases three, pyrheliometers, one having its sensitive plane horizontal and the others having this plane vertical and facing either east, south or west. The time-solar radiation intensity curves for the other walls were then determined trigonometrically, a practice which earlier Laboratory work has proved satisfactory.

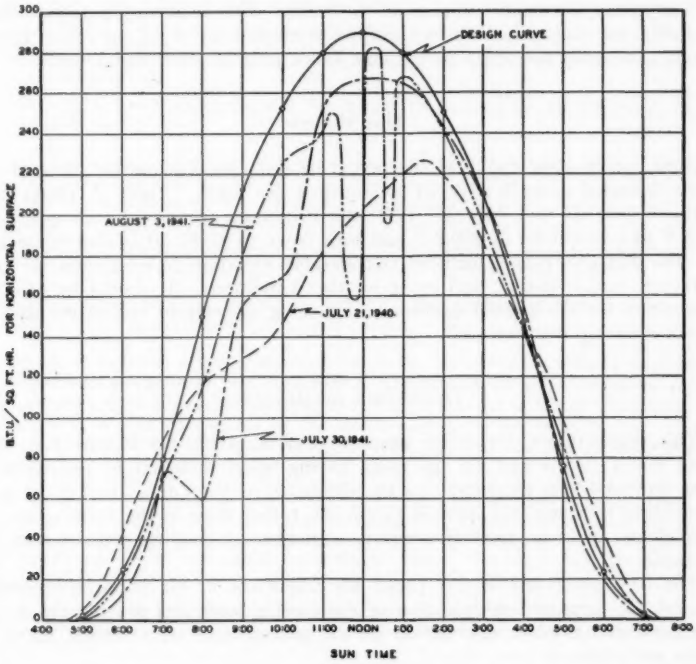

FIG. 8. DESIGN SOLAR RADIATION ON A HORIZONTAL SURFACE FOR AUGUST 1 AND THAT OBSERVED FOR DIFFERENT TEST DAYS

HEAT GAIN RELATIONSHIPS FOR DIFFERENT ORIENTATIONS AND TYPES OF CONSTRUCTION

Since a number of the types of construction studied had only one orientation, as indicated in Table 1, and, further, since a number of these types of construction, particularly those containing insulation, were only studied on a few satisfactory days and then late in the season, it was necessary to rely upon relationships found for certain walls tested with different orientations and on data collected on several good days to arrive at the relationships on which the entire series of curves in Figs. 1 to 6 are based.

The 12-in. brick and the brick and tile walls were the only walls studied

with four orientations; the brick and frame wall was studied with east, south and west exposures only. From the results obtained from these walls, it was found that with minor variations the heat flows through the east, south and north walls were 83, 75, and 46 per cent, respectively, of that for the west wall.

The time relationship between the minimum and maximum heat flows through the several walls with the four orientations was fixed by averaging the results for certain days for the few walls studied with all four orientations and applying this relationship to all the walls. This relationship applies satisfactorily for walls which do not differ too much from those on which it was based. It would not apply to extreme cases such as sheet metal walls.

HOT WAVES

Both during 1940 and 1941 the period of test observations for most of the walls included several days of a so-called hot spell. Table 3 shows that between July 23 and July 31, 1941, the maximum temperature ranged from 88.5 F to a maximum of 100.2 F and back down to 87 F. It is also interesting to note that this period included considerable variation in maximum air temperature, sun, radiation, and other weather conditions. It should be of some interest in the design and operation of cooling systems to record the data in Table 3 during this hot spell.

DISCUSSION OF RESULTS

This report offers, for design purposes, curves giving the 24-hour heat gain rates for 11 walls and for 18 roofs having wide variation in resistance to heat flow and heat capacity. As the character of the curves in Figs. 1 to 6 themselves indicates, the data in this form, rather than in the form of simple coefficients, must be used if accurate estimates of cooling requirements are desired.

The data presented in this paper are important as an initial contribution toward the accurate determination of the cooling load, and provide the design engineer with needed information on the performance of a typical series of walls and roofs.

Further study of the data is necessary to determine how the observed heat flows compare with the theory relating heat flow to the orientation, heat capacity, transmittance coefficient, and absorptivity of solar radiation by the outside surface of the structure. There is also a need for determination of the effect of re-radiation from adjacent building structures. While it is hardly feasible to establish curves for all types of walls and roofs encountered, it may be possible to include in one chart for each orientation curves for a sufficient number of types of construction, including a wide range in the resistance to heat flow and heat capacity, to permit interpolation for other similar constructions. Data are also desirable for other orientations such as northeast, southeast, northwest and southwest exposures and for other roof slopes. This study suggests the need for additional work to help establish the maximum expected rate of heat flow, particularly with reference to the probable coincidental occurrence of both design solar radiation and design outside temperature.

DISCUSSION

W. F. FAIR, JR., Pittsburgh, Pa. (WRITTEN): First I should like to present six comments concerning this paper:

1. Roof No. 6, although being described in Table 2 as being similar to roof No. 3 has tabulated characteristics of roof No. 4 (which is roof No. 3 insulated); in Fig. 6, Curve 6 is described as an insulated roof, gravel surfaced.
2. Roofs Nos. 3 *vs.* 1, and 4 *vs.* 2, and 16 *vs.* 18 show the beneficial effect of addition of slag.
3. Roofs Nos. 5 *vs.* 3 (and in turn *vs.* 1), and Nos. 15 *vs.* 17 illustrate the improvement due to double pitch and slag.
4. Roofs Nos. 6 *vs.* 4 indicate almost negligible superiority of gravel over slag (provided amounts used and thicknesses are the same—U and Heat Cap. are equivalent).
5. Roofs Nos. 7, 5 and 3 indicate that a second pitch application may be approximately equivalent to 2 in. of concrete in the base.
6. If roofs Nos. 7 and 11 were given a double pitch application their curves might drop down to the peaks for insulated materials; and if Nos. 8 and 12 were given double pitch coatings their curves might be expected to be even flatter and lower.

These results emphasize the value of applying double pitch and slag coatings to roofs. It seems probable that a triple application, if practicable, would give even better results, or permit the use of smaller thickness of base material for approximately the same heat flow characteristics; an additional application to an easier means to effect improved heat flow characteristics (together with simultaneous increase in resistance to weathering) than by interior constructional methods.

DR. HOUGHTEN: To reply adequately to all of the questions involved would take a little time, but I believe they can be generalized.

Mr. Fair brought out a typographical error in his first comment and the particular roof mentioned should have been indicated as gravel covered with insulation. One of the most surprising results from the roof study was the apparent magnitude of the resistance to heat flow added by the waterproofing portion of a number of the roofs. It is true that the waterproofing member on the 2-in. concrete roof has about the same effect as the 2-in. concrete itself.

One factor accounting for the discrepancy is the fact that the waterproofing considered in earlier GUIDE values was very thin. The waterproofing members used in these tests were built up to from a quarter of an inch to three-eighths. Apparently the gravel had just about the same effect as slag of the same thickness. Contrary to the usual conception the slag had considerable effect. Probably due, not so much to the added resistance to heat flow, as to the added heat capacity.

JOHN EVERETTS, JR., Pittsburgh, Pa.: By reference to Table 3, giving the weather and heat data during the hot wave, there are some very interesting details reported. It seems that the design curve data for heat gain through walls under various conditions as compared to the data in Table 3 are a little high. Apparently this is due to the conservative feeling on the part of the Laboratory in that they have considered maximum solar radiation and maximum temperature to be coincidental.

From the work of the Research Technical Advisory Committee on Weather Design Conditions we do not feel that a coincident condition of maximum sun effect and temperature is possible, inasmuch as the maximum solar radiation would be impeded by haze or fog or dust particles which would tend to settle during maximum temperature conditions due to the lack of circulation of atmospheric air. Therefore, we feel that the design values as obtained from the curves in this paper will give greater results than required.

By taking the design conditions reported we find that for the east wall as given in the table, which shows a maximum heat transfer of 5 Btu per square foot per hour, the 1940 and 1941 editions of THE GUIDE would show equivalent heat transfer values by calculation of 16.9 Btu or about three times as much as the value given in Table 3, and also from the curve which indicates 6 Btu.

On the west wall we show a comparison of the Guide recommendation value of 9 as against the curve data of 7.5. On the south wall a Guide value of 4.5 against

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the curve value of 3.5, and similar wall comparisons could be made. This means that if we take the example as given in the Chapter on Cooling Load of THE GUIDE we find that in the 1940 GUIDE the particular little store application problem which has been used in THE GUIDE from 1932-1940, without much change in physical dimensions, started out with a load of 9.6 tons. That same store now only requires 7.3 tons, or 7.5-ton unit, or a 25 per cent reduction to the owner in the selection of equipment.

In view of the analysis which the ASHVE Research Laboratory has done in determining heat flow curves through building walls, which has reduced the values in our calculated measurement, or the application of comfort air conditioning, it has definitely brought out improvement in a design application and engineering of such installations. We would like to see additional information of other types of walls and roofs investigated at the Society's Laboratory to complete this program as soon as possible.

C. M. ASHLEY, Syracuse, N. Y.: I would like to suggest that this paper be considered as a part of a continuation program, or in other words, somewhat in the nature of a progress report. The Laboratory work is still being carried on and quite a lot of work remains to be done in order to finish the job and provide all the data that are needed by the air conditioning industry. The Research Technical Advisory Committee on Cooling Load and Summer Air Conditioning felt that these data were of such value that it fully justified the publication of the report at the present time rather than hold it until such time as we had obtained complete information.

It is hoped that over the period of the next few months further work can be done on this subject which is of an analytical character. First it is hoped that the data which have already been obtained can be studied from a mathematical point of view, so that a formula can be developed to predict the performance of a variety of other types of walls and roofs.

Further than that it is hoped that we can obtain additional data, such as the actual heat transmission through the wall, the amount of absorption on the surface of a wall and the collection of weather data to determine whether the solar radiation assumed and the wet and dry-bulb temperature conditions assumed for the design curve are actually coexistent or not.

DR. PASCHKIS: I believe that Dr. Houghten and his co-workers did a marvelous piece of work as reported in this paper, but there are two things that come to my mind. One is that the interference of the outside wind is not mentioned and I believe the experiments which were presented in my paper show that this factor has a great influence on the wall. Consequently, I feel that it is important to carry on the experiment in such a way that a mathematical formula can be devised to select the heat flow through various types of walls.

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No. 1196

COMFORT WITH SUMMER AIR CONDITIONING

Report of the ASHVE Research Technical Advisory Committee on Sensations of Comfort: Thomas Chester, *Chairman;* N. D. Adams, C. R. Bellamy, G. D. Fife, E. P. Heckel, Dr. W. J. McConnell, F. C. McIntosh, A. B. Newton, B. F. Raber and C. Tasker

A GENERATION ago it was considered that ideal indoor atmospheric conditions should be similar to those of a perfect day in June. This was not very precise but the idea was good. Nature does provide such delightful days toward the end of Spring in the northern portion of the United States and earlier in the southern regions.

Human comfort is promoted in winter when the aerial envelope in contact with the skin below the clothing is easily maintained in a sub-tropical condition.

In summer the metabolic heat which can be lost by radiation and by convection is less, so it is necessary for the body to lose more heat by the evaporation of perspiration under natural atmospheric conditions. In hot and humid climates and also in densely-occupied enclosures in temperate climates in summer, the water vapor content of the atmosphere is too high for comfort. Under such conditions it is usually necessary to reduce both the dry-bulb and dew-point temperatures in order to assist the human body to dispose of heat by convection and by evaporation. It should also be noted that when the dry-bulb temperature is lowered, this results in a reduction of the average temperature of the interior surfaces of an enclosure and consequently more heat can be lost by the body by radiation.

It may be considered that the body's primary means of maintaining its required temperature is its regulated disposal of heat by radiation and by convection, and that heat loss by the evaporation of perspiration is a secondary method. It seems rational to assume the complete environment, atmospheric and physical, should be such that the body can freely use all its methods of heat disposal, and that it should not be compelled by environmental conditions to make undue use of one particular method, i.e., the evaporation of perspiration to the exclusion of the others.

An average adult in a state of little activity must lose heat at the rate of approximately 400 Btu per hour, both in summer and winter. Comfort is attained when this amount of heat can be lost without any sensation either of cold or heat.

Presented at the 48th Annual Meeting of the American Society of Heating and Ventilating Engineers, Philadelphia, Pa., January, 1942.

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In the early days of air conditioning for comfort in summer, the chief aim in most cases was to remove sensible and latent heat from the indoor atmosphere; usually too much of the former and too little of the latter, with the result that the air felt cold and noticeably damp. Even today individuals who are susceptible to colds and pulmonary troubles are advised by their physicians to avoid air conditioned theaters and restaurants.

During the past decade a great deal of research work has been carried on in order to ascertain indoor atmospheric conditions conducive to optimum comfort. Studies have been made over a considerable geographical range embracing Toronto, Minneapolis, Pittsburgh, New York, Washington, and San Antonio. The feelings and reactions to summer air conditioning of very many individuals have been carefully noted. The subjects included men and women, old and young, and attention was given to the clothing worn.

This resulted in the presentation to the Society of many papers containing a wealth of detailed information, with ably deduced summaries and conclusions, and disclosing a general conformity which amply warrants reliability.

During 1940 the Committee on Research requested the Technical Advisory Committee on Sensations of Comfort to review eleven papers and in consonance with them to make recommendations concerning indoor effective temperatures for summer occupancies. The papers given this consideration were:

1. ASHVE RESEARCH REPORT No. 1035—Comfort Standards for Summer Air Conditioning, by F. C. Houghten and Carl Gutberlet. (ASHVE TRANSACTIONS, Vol. 42, 1936.)
2. ASHVE RESEARCH REPORT No. 1055—Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker and Carl Gutberlet. (ASHVE TRANSACTIONS, Vol. 43, 1937.)
3. Cooling Requirements for Summer Comfort Air Conditioning in Toronto, by C. Tasker. (ASHVE TRANSACTIONS, Vol. 44, 1938.)
4. ASHVE RESEARCH REPORT No. 1088—Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley. (ASHVE TRANSACTIONS, Vol. 44, 1938.)
5. ASHVE RESEARCH REPORT No. 1101—Seasonal Variations in Effective Temperature Requirements, by F. E. Giesecke and W. H. Badgett. (ASHVE TRANSACTIONS, Vol. 44, 1938.)
6. ASHVE RESEARCH REPORT No. 1102—Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices, by A. B. Newton, F. C. Houghten, Carl Gutberlet, R. W. Qualley and M. C. W. Tomlinson. (ASHVE TRANSACTIONS, Vol. 44, 1938.)
7. ASHVE RESEARCH REPORT No. 1103—General Reactions of 274 Office Workers to Summer Cooling and Air Conditioning by F. C. Houghten, A. B. Newton, R. W. Qualley and Edward Witkowski. (ASHVE TRANSACTIONS, Vol. 44, 1938.)
8. ASHVE RESEARCH REPORT No. 1127—Reactions of Office Workers to Air Conditioning in South Texas, by A. J. Rummel, F. E. Giesecke, W. H. Badgett and A. T. Moses. (ASHVE TRANSACTIONS, Vol. 45, 1939.)
9. ASHVE RESEARCH REPORT No. 1136—Summer Cooling Requirements in Washington, D. C., and Other Metropolitan Districts, by F. C. Houghten, Carl Gutberlet and Albert A. Rosenberg. (ASHVE TRANSACTIONS, Vol. 45, 1939.)
10. ASHVE RESEARCH REPORT No. 1153—Seasonal Variation in Reactions to Hot Atmospheres, by F. C. Houghten, A. A. Rosenberg and M. B. Ferderber. (ASHVE TRANSACTIONS, Vol. 46, 1940.)
11. Reactions of 745 Clerks to Summer Air Conditioning, by W. J. McConnell and M. Spiegelman. (ASHVE TRANSACTIONS, Vol. 46, 1940.)

Other papers of pertinent interest to this subject, which have been published recently by the Society, include:

1. ASHVE RESEARCH REPORT No. 1079—Physiologic Response of Man to Environ-

mental Temperature, by F. K. Hick, R. W. Keeton and Nathaniel Glickman. (ASHVE TRANSACTIONS, Vol. 44, 1938.)

2. Recent Advances in Physiological Knowledge and their Bearing on Ventilation Practice. Report of ASHVE Research Technical Advisory Committee on Physiological Reactions; C.-E. A. Winslow, *Chairman*. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

3. ASHVE RESEARCH REPORT No. 1111—Air Conditioning Requirements of an Operating Room and Recovery Ward, by F. C. Houghten and W. Leigh Cook, Jr. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

4. ASHVE RESEARCH REPORT No. 1151—The Peripheral Type of Circulatory Failure in Experimental Heat Exhaustion—The Role of Posture, by Robert W. Keeton, Ford K. Hick, Nathaniel Glickman and M. M. Montgomery. (ASHVE TRANSACTIONS, Vol. 46, 1940.)

5. ASHVE RESEARCH REPORT No. 1174—Comfort Requirements for Low Humidity Air Conditioning, by F. C. Houghten, H. T. Olson and S. B. Gunst. (ASHVE TRANSACTIONS, Vol. 47, 1941.)

6. ASHVE RESEARCH REPORT No. 1175—The Influence of Physiological Research on Comfort Requirements, by Robert W. Keeton, Ford K. Hick, Nathaniel Glickman and M. M. Montgomery. (ASHVE TRANSACTIONS, Vol. 47, 1941.)

Resulting from a review of the first eleven papers listed, it has been decided to recommend that the United States be divided into four zones from north to south, embracing a range of recommended indoor effective temperatures for design from 70 to 73 deg. Fig. 1 shows the four zones and the corresponding recommended indoor effective temperatures.

Any idea that finality has been reached is expressly disclaimed, as there still remains a wide field for research. The recommendations apply to buildings approximately similar in design and construction to those in which the comfort investigations were made. In general these were of average modern type with normal walls and windows. Radical departures in design can involve variations in effective temperatures conducive to comfort.

Taking an extreme case, it is axiomatic that in an uninsulated galvanized iron structure a lower effective temperature would be desirable, in order to compensate for the effect of inward radiation from the boundary surfaces subjected to solar heat. Maintenance of the indoor effective temperature indicated for the geographical location of such a building would not be suitable, even with the solar heat gain properly computed in the required cooling load.

It will be obvious that with a rectangular building with long north and south walls practically all glass, with short windowless east and west walls, and with a central partition wall running east to west, comfort conditions would be dissimilar in the two enclosures even if the same effective temperatures were maintained. The difference would be very noticeable with unprotected windows, and perceptible even if awnings were used, due to the difference in heat disposal by radiation from the occupants of the two rooms. In a paper[1] previously published by the Society, it is reported that one of the subjects stated in unscientific but unequivocal language:

> Air conditioning is okey, except along the south windows—when the sun shines in here it is hotter than outside. I suggest that you turn on more cold along the windows.

Theaters are windowless and usually have false ceilings with roof air spaces which afford protection against solar radiation. On the other hand, they

[1] ASHVE RESEARCH REPORT No. 1103—General Reactions of 274 Office Workers to Summer Cooling and Air Conditioning, by F. C. Houghten, A. B. Newton, R. W. Qualley and Edward Witkowski. (ASHVE TRANSACTIONS, Vol. 44, 1938.)

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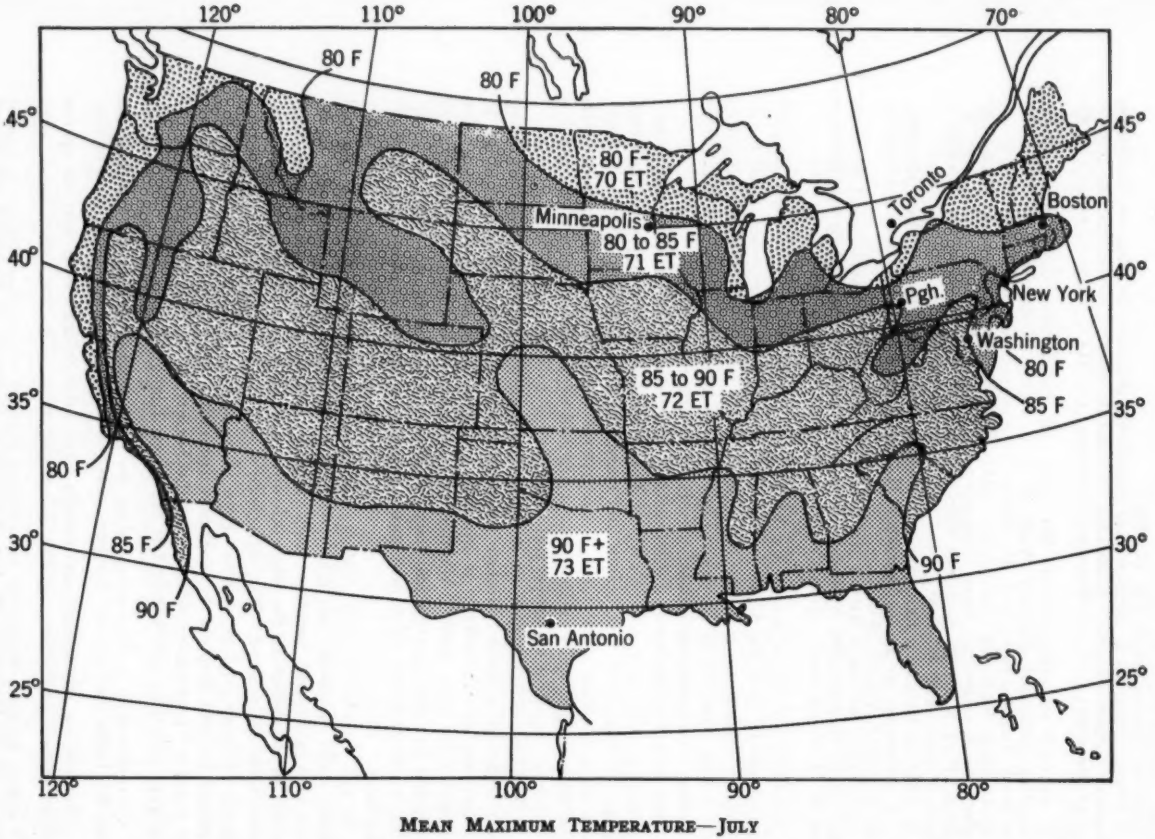

MEAN MAXIMUM TEMPERATURE—JULY

FIG. 1. THE ABOVE EFFECTIVE TEMPERATURES APPLY TO BUILDINGS OF COMMON CONSTRUCTION WHEREIN NO UNUSUALLY LARGE AREAS ARE AFFECTED BY SOLAR RADIATION. THEY ARE SUITED TO LONG PERIODS OF OCCUPANCY BY PEOPLE IN A STATE OF VERY LITTLE ACTIVITY OR ENGAGED IN LIGHT SEDENTARY OR CLERICAL WORK. AIR VELOCITIES 15 TO 25 FPM.

contain many more persons per thousand square feet of floor area than office buildings. Closely packed occupants do not lose much heat laterally by radiation and loss by convection is less than in more sparsely occupied enclosures. Consequently, the selection of suitable effective temperatures for theaters offers scope for research.

Compensatory action for warm boundary surfaces can be produced by lowering the dry-bulb temperature while still maintaining the same effective temperature, but this of course involves increased relative humidity, and more satisfactory results can be obtained by lowering the effective temperature and maintaining a satisfactory relative humidity.

The recommended effective temperatures for the four zones shown by the map are for long occupancy periods. In places such as department stores and banks the comfort requirements of customers and employees differ. Individuals entering such air conditioned enclosures for short visits find an effective temperature best suited to the employees to be too cold for comfort. The chief cause of the cold shock upon entering is the quick cooling of perspiration laden clothing due to a hot and humid outdoor atmosphere. Evaporation tends to lower moist garments to the wet-bulb temperature of the conditioned air, which is considerably below the dry-bulb temperature.

For instance at 73 deg ET with 50 per cent relative humidity, the dry-bulb temperature is approximately 79 F and the wet-bulb temperature 66 F. It is not surprising that a pronounced chilling effect is produced but what can be done to counteract it is not definitely known. In cases where most persons enter an air conditioned enclosure at the same time, it would seem that initially the effective temperature could be about a mean between the outdoor effective temperature and that considered desirable for indoors, with gradual reduction to the required air condition in 20 to 30 min. Although experiments which have been made along this line have not been very successful, this need not dispose of the method and further investigation seems warranted. It must be admitted that in very many cases attempts to provide comfort in summer by means of air conditioning result in an initial feeling of pronounced discomfort.

As regards leaving shock it is generally considered that going into the hot and humid outdoor atmosphere from an air conditioned space is very unpleasant. Some engineers are of the opinion that this is due to the lag of the sweat glands in producing the necessary increased exudation of perspiration, and that to some extent this may result from the higher vapor pressure outdoors. Although evaporative cooling is sequentially the human body's secondary means of temperature control by heat disposal, it is much more potent than cooling by radiation and convection. The only method in sight at present for reducing leaving shock is gradual raising of the indoor effective temperature and this seems restricted to enclosures where all the occupants leave at about the same time. Maintenance of higher vapor pressures indoors is not feasible because this would involve higher dewpoints and increased relative humidities.

In the paper[2] previously mentioned, one individual is reported as having stated:

After leaving the air conditioned office and entering the warm unconditioned places,

[2] Loc. Cit., Note 1.

LARGE AREAS ARE AFFECTED BY SOLAR RADIATION. ... STATE OF VERY LITTLE ACTIVITY OR ENGAGED IN LIGHT SEDENTARY OR CLERICAL WORK. AIR VELOCITIES 15 TO 25 FPM.

perspiration began to flow freely, and for a short time a sort of dizzy feeling would be felt, but would soon wear off.

This was evidently an extreme case because in this same paper under the heading Discussion of Results the following appears:

. . . Better than 35 per cent were frank in stating that they felt too warm for a while after leaving the air conditioned space. However, the duration of the hot reaction was in most cases not great, and no importance was placed on it. . . .

In another paper,[3] it is stated:

The warm shock that is felt by persons leaving an air conditioned building is a sensation that is not easily analyzed. It is needless to say that any condition is emphasized by contrast, but that should not necessarily mean that the condition is harmful because of the contrast . . . Practically the entire group expressed a feeling of discomfort immediately after leaving the building.

In general the investigations did not disclose that any harmful results ensued from leaving shock. From a practical standpoint it does not seem necessary for air conditioning engineers to be much concerned about this matter, as after all any discomfort felt upon leaving an air conditioned enclosure carries with it an appreciation of the comfort felt therein.

On the other hand entrance shock can be harmful to persons in delicate health and the public unquestionably reacts unfavorably to cold chills and feelings of clamminess, so research work in this direction seems worth while.

THE GUIDE defines *Comfort Zone* (average) as the range of effective temperatures over which the majority of adults feel comfortable. Comfort does not suddenly diminish on each side of the optimum condition when the air contains either more or less total heat than is regarded as ideal. As shown by the various papers, comfort curves are parabolic, whether pertaining to the sensations of one individual under varied atmospheric conditions, or to the reactions of large groups of people to the same effective temperature. Ideal conditions are at the apices of such curves. Starting at the base of a curve on the cold side, seven primary and perceptible variations in comfort can be indicated graphically, as follows:

(1) Cold. 9 deg ET below optimum.
(2) Too cool. 6 deg ET below optimum.
(3) Comfortably cool. 3 deg ET below optimum.
(4) Ideal comfort. Optimum effective temperature.
(5) Comfortably warm. 3 deg ET above optimum.
(6) Too warm. 6 deg ET above optimum.
(7) Hot. 9 deg ET above optimum.

Individuals vary considerably in their opinions of the effective temperatures conducive to comfort. A great deal of information is given in the papers listed which it does not seem necessary to reprint in this digest.

It seems to have been proven that women prefer somewhat higher effective temperatures than men and that people over forty like higher temperatures than those under forty. This indicates the impossibility of establishing effective temperatures which are ideal for all people and all that can be done is to establish standards which will be objectionable to the smallest number. It is perhaps reasonable to attribute the difference in opinion of men and women

[3] ASHVE REPORT No. 1127—Reactions of Office Workers to Air Conditioning in South Texas, by A. J. Rummel, F. E. Giesecke, W. H. Badgett and A. T. Moses. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

in this matter to differences in clothing. It is rational to suspect that those men who feel somewhat too warm in an indoor atmosphere which suits the large majority of men and women are wearing heavier attire.

As previously stated the recommended effective temperatures apply to enclosures of common construction not unduly affected by solar radiation. They are suited to long periods of occupancy by people engaged in light sedentary work. Lower effective temperatures are necessary for persons engaged in occupations involving greater activity, as heat output increases with muscular effort.

The recommended effective temperatures also apply to enclosures in which the average air velocity is from 15 to 25 fpm. Perhaps in the future it will be possible to provide higher air velocities and thus go a little way in imitating nature, as the average air movement outdoors is in excess of 500 fpm. With present types of building construction it is not feasible in most cases to exceed the air velocities stated, without causing objectional drafts.

The earliest recorded case of the employment of mechanical refrigeration in cooling air for comfort, was that of an ammonia system installed by a British firm at a Rajah's palace in India in 1887. In that country cooling by means of air movement has been in use from time immemorial. Manually operated punkahs or air beaters have generally been superseded by motor driven ceiling fans where electricity is available. In offices the overhead fans discharge air towards the ceiling and this provides good air movement without disturbance of papers on desks.

Desk fans and ceiling fans are still in wide use in the United States. In the drier regions their use along with evaporative cooling goes a long way in the alleviation of hot weather discomfort. Generous air velocities can be arranged as under such conditions drafts are not unpleasant, the effect being more or less similar to that of a breeze in the open air. Air conditioning by means of refrigerated air is in a different category, as currents of air introduced at 15 F or more below the maintained room temperature can be very objectionable.

In many tropical countries it is customary to provide buildings with wide verandahs to keep out the sun and rain, and to allow free air movement by means of large open windows. In air conditioning work the fascination of mechanical contrivances has almost obscured the old and inexpensive efficacy of removing heat from the skin by means of air circulation.

The question of relative humidity must receive consideration. The Comfort Chart developed in the Society's Laboratory and reproduced in THE GUIDE has stood very well the test of time. Recent investigations have disclosed its substantial accuracy for comparatively low relative humidities—in the neighborhood of 30 per cent. Although as a matter of scientific interest it was desirable to study comfort reactions with relative humidities up to 100 per cent, no one would think of using them for ordinary installations. The Comfort Chart shows a shaded area indicating a desirable relative humidity range of from 30 to 70 per cent. It is recommended that the relative humidity for summer comfort should not be in excess of 60 per cent and that more satisfactory results can be obtained with lower relative humidities.

As regards to the four different effective temperatures shown by the map, human adaptation to climatic conditions is generally accepted and it seems reasonable to believe that an individual who lives in the lowlands of the deep

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south where sugar cane is grown is more inured to heat and humidity than a person who lives near the Canadian border.

This is backed up by experiences in other parts of the world. It was found that summer comfort was obtained at much higher than 71 deg ET for foreigners residing in Shanghai, where climatic conditions are more severe as regards heat and humidity than those of the Gulf states in this country. The London School of Hygiene and Tropical Medicine recommends 85 F dry-bulb and 72 F wet-bulb for Nigeria, and this represents 78.2 deg ET, which is well outside the shaded area shown on the Comfort Chart.

For ships which travel the tropic oceans, a suitable condition of the outboard atmosphere assumed as a basis for design, which occasionally will be exceeded, is 90 F dry-bulb, 84 F wet-bulb, 80 per cent relative humidity, 82.5 F dew-point, and 11.95 grains of water vapor per cubic foot of moist air. For this condition the effective temperature is 86 deg. The high humidity is due to the fact that the ocean surface is at 82 to 85 F, with water temperatures in excess of these in ports and in narrow seas.

Under such climatic conditions it is suitable to maintain in the fully occupied air conditioned spaces 82.5 F dry-bulb, 69 F wet-bulb, 50 per cent relative humidity, 62 F dew-point, and 5.96 grains of water vapor per cubic foot of conditioned air. The effective temperature for this condition is 76 F, which is well in excess of the standard recommended for the southernmost portion of the United States. Any appreciably lower effective temperature than stated would be too cold on the tropic oceans, because the passengers spend a great deal of time on deck in the open air and tend to become acclimated to its high total heat content.

The Comfort Chart shows an optimum of 66 deg ET for winter and 71 deg ET for summer. This is indicative of adaptation of the human body to climatic conditions, so to summarize the foregoing it seems rational to provide different optimum indoor effective temperatures in different portions of the United States.

In most places heat intensity is governed by the mid-day elevation of the sun above the horizon, so latitude is the chief determining factor. As regards altitude, the temperature of the atmosphere falls somewhat more than 5 F for each increase of 1,000 ft in elevation; the effect being modified by a slightly increased temperature of surfaces exposed to direct rays of the sun. The direction of prevailing winds also influences climates, as a wind is an atmospheric drift. In summer in this country the wind is mostly from the southwest. In the northern hemisphere sustained north winds result in lower air temperatures. In seaboard regions ocean currents also affect climates. On the west coast the current from the north tempers summer heat intensity, and on the east coast the Gulf Stream moderates the rigors of winter as far north as Cape Hatteras.

The endless procession of the seasons is due to the inclination of the earth's axis to the plane of its orbit around the sun. This causes the elevation of the sun at noon to be 47 deg higher on June 21 than on December 21 for regions in the northern hemisphere. Disregarding the reductions consequent to varying distances of penetration of the atmosphere, solar heat received per square foot of horizontal surface varies with the sine of the angle of the sun's elevation. It should be noted that places in high latitudes have more hours of sunlight per day in summer than places nearer the equator, and this

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introduces a variation factor. Maximum temperatures are reached in most regions about four weeks after maximum solar elevation, due to accumulation of heat in the atmosphere. Temperature drop during the night, due to radiation to interstellar space, is least when the sky is cloudy and greater when there are no clouds. Although entirely outside the field of air conditioning and in no way a substitute therefor, discomfort caused by summer heat can be somewhat alleviated by means of copious ventilation with relatively cool night air.

The effective temperatures shown in Fig. 1 are based on the design data given in the 1941 edition of THE GUIDE (p. 126). Design dry-bulb and wet-bulb temperatures are not maxima and consequently when peak summer weather conditions are experienced, the recommended indoor effective temperatures will be exceeded. Refrigerating plants are usually figured for design conditions and beyond their maximum capacity indoor dry-bulb and wet-bulb temperatures float with the total heat load. This is sometimes aggravated by decreased refrigerating effect in hot and humid weather, especially when atmospheric condensers are used.

With indoor effective temperatures as shown in Fig. 1, accurate thermal appraisals with recently available instruments, including globe thermometers, would disclose a greater difference between the northern and southern zones than is denoted by a difference of 3 deg in effective temperature. When it is considered that in one case the outside atmosphere is around 80 F and in the other case above 90 F it will be realized that room boundary internal surface temperatures will be somewhat more in excess of maintained indoor dry-bulb temperatures in the south than in the north. The differential which has a direct effect on comfort due to the lessened heat disposal from the occupants by radiation, is increased owing to the fact that the walls of buildings are usually thinner in the south than in the north.

As is well known, air conditioning is largely used in the Government buildings in Washington, D. C. Recent information concerning them, although not appearing in the papers reviewed, will be of interest.

For various outside dry-bulb temperatures the logs for July and August, 1940, indicate that the following effective temperatures were satisfactory for the Senate Chamber:

OUTSIDE DEG F	INSIDE EFFECTIVE TEMP.
Under 76	70.5
Between 76 and 80	71.0
Between 81 and 85	71.4
Between 86 and 90	71.8
Between 91 and 95	72.7

The foregoing effective temperatures are averages of eight readings for each group; each reading taken on a different day. The readings are scattered through the two months, with the exception of the last group which is confined to a period of about three weeks during the last of July. A study of the logs in sequence reveals a trend which is not apparent from the table given. This is, that for sudden changes in outside temperatures very little or

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no change in inside effective temperature is desirable. However, for a gradual increase in outside temperatures over a few days, it is desirable to increase the effective temperatures indoors.

This checks with the theory that the human body adapts itself to climatic conditions. After a pronounced heat wave in New York City which commenced July 19, 1940, and continued for 12 days, it was observed that when it abated somewhat, comfort was experienced in an un-air-conditioned room in which the atmosphere was at 85 F dry-bulb, 73 F wet-bulb, 56 per cent relative humidity, with interior surface temperature of the walls at 82 F. The equivalent effective temperature of 78.6 deg was conducive to comfort after tropical climatic conditions lasting 12 days.

Reverting to Washington, endeavor is made there to maintain dry-bulb temperatures of from 76 to 78 F with 50 per cent relative humidity throughout the buildings on Capitol Hill during the summer. With very few exceptions the room temperatures may be varied at will by the occupants by resetting the individual room thermostats. It has been found that people from southern states, older people, and women, require in general a higher temperature than others. From experience those in charge are strongly opposed to indiscriminate changing of indoor conditions to comply with any theoretical desirable relation between inside and outside temperatures, for the particular buildings under their supervision. It has been ascertained that too many variables are involved, such as the type of occupants, the length of occupancy, the rate of outdoor temperature change, etc. The logs for the Senate Chamber do not show the desirability of maintaining somewhat lower than normal effective temperatures to compensate for counter radiation between the occupants. It is thought that this may be explained by the fact that the spacing of the occupants is much more generous than in an average theater, and also because the occupants are well above the average age. In the Capitol buildings the length of occupancy is from 3 to 9 hours, and it is considered important to maintain conditions desired by the regular occupants, without regard to the reactions of transients. It is held that the matter of entrance shock has been over-emphasized by others, as it is found of no importance. However, during the summer many of the entrance halls have higher temperatures than the interior rooms, and this is regarded as desirable. It is recognized that entrance shock is reduced by means of the warmer entrance halls.

In winter, maximum comfort is obtained by maintaining a 68 deg ET, which is 2 deg in excess of the optimum indicated by THE GUIDE, but it is thought that this again can be attributed to the fact that the occupants mainly considered are of more than average age.

This paper may be regarded somewhat as an interim report, as there is still a good deal to be learned concerning the interplay of radiation as affecting comfort in summer, between the occupants of air conditioned buildings and their physical environments. It is desirable that research be made in order to gain knowledge of important factors which at present are obscure or unknown. The Technical Advisory Committee on Radiation and Comfort has this subject under consideration and research work regarding it is now being carried on at the University of California.

In conclusion it must be stated that the four zonal effective temperatures shown in Fig. 1 should not be considered as inflexible requirements in all

cases. Economic aspects have to be regarded and where limited financial resources prevent the installation of equipment capable of producing ideal comfort in summer, the aim should be to alleviate discomfort as far as funds permit.

DISCUSSION

DR. R. W. KEETON, Chicago, Ill. (WRITTEN): It was previously pointed out [4] that comfort is an affective state produced in the individuals by sensations of heat and cold induced by stimuli arising from the skin. In attempting to gain knowledge of atmospheric conditions which produce comfort, investigators have used in some instances terms which describe sensations (*cold, cool, slightly warm, warm*) and in other instances terms which describe the affective state (*very unpleasant, unpleasant, indifferent, pleasant, and very pleasant*). It is obvious that when the individual comes into a state of neutrality and is unaffected by his environment, he is comfortable. This separation of sensations from their effects on the individual may at first seem unnecessary, but the significance of the distinction should become apparent later.

Gagge, Winslow and Herrington [5] have found that in the nude subject the skin temperature ranges for pleasantness lie between 92.5 F and 91 F. The zone of unpleasantness was found on either side when the skin temperatures reached values of 87.1 F and 94.5 F. In clothed subjects [6] these investigators found again the sensations of pleasantness to be associated with skin temperatures between 90.8 F and 92.4 F. The zones of unpleasantness were reached with skin temperatures of 87 F and 94.7 F. So they concluded that comfort was best correlated objectively with skin temperature. It has come to be appreciated that a temperature that is comfortable to men is, as a rule, too cool for women and one that is acceptable to women is too warm for men. A physiological basis for this difference has been suggested by Hardy and Du Bois,[7] in that women produce less heat in the warmer environments, and in the cooler zones their thicker layer of fat in the subcutaneous tissues furnishes an insulation against cold. More recently Yaglou and Messer [8] have attributed these differences in comfort requirements solely to the type of clothes worn. Thus, when the men were dressed in the women's clothing, they behaved as women in their comfort requirements and chose a warmer atmosphere. In a similar manner, when women were dressed in the clothing of men they preferred a cooler atmosphere. They found that comfort in all but a few subjects was associated with mean skin temperatures between 91.5 F and 93.5 F. The exposed skin temperature was a poor index of comfort and the external temperatures of the clothing bore no correlation to comfort. It should be understood that these experiments apply to subjects who were resting or engaged in minimal work. They were in essentially *steady states* from a physiological viewpoint. The results would not necessarily hold for a subject at work whose heat production was elevated or for subjects whose peripheral vessels were undergoing violent fluctuations necessitated by great thermal exchanges.

If these experiments are confirmed by other investigators, they will explain satisfactorily the difference in comfort standards of the sexes, but they will not prove

[4] ASHVE RESEARCH REPORT No. 1175—The Influence of Physiological Research on Comfort Requirements. R. W. Keeton, F. K. Hick, N. Glickman and M. M. Montgomery. (ASHVE TRANSACTIONS, Vol. 47, 1941.)

[5] The Influence of Clothing on the Physiological Reactions on the Human Body to Varying Environmental Temperatures, A. P. Gagge, C.-E. A. Winslow and L. P. Herrington. (*American Journal Physiology* 124: No. 1, 1938.)

[6] Loc. Cit. see Note 3.

[7] Difference Between Men and Women in Their Response to Heat and Cold, J. D. Hardy and E. F. Du Bois. *Proceedings National Academy Science*, 26: 389, 1940.)

[8] The Importance of Clothing in Air Conditioning, C. P. Yaglou and A. Messer. (*Journal American Medical Association*, 117: 1261, 1941.)

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of much practical value in persuading men and women to adjust their clothing habits for the benefit of the engineer. However, these experiments would give support to the committee's suggestions that a higher effective temperature be used as a standard in the southern part of the United States, and a lower one in the northern part. It is well understood [9] that the people of the South habitually dress in lighter clothes than those in the North. Hence the skin temperatures of the Southerner would be held at the comfort range more readily by a higher effective temperature of the environment. In the North the conditions would be reversed.

Many experiments have been reported which indicate that acclimatization plays an important role in establishing the limits of comfort zones. Some are open to criticism. However, there seems to be little doubt that a correlation exists. The factors responsible for the correlation are not clear. Yaglou and Drinker [10] have noted that the American people prefer indoor temperatures 8 F higher than those reported by Vernon as comfortable to the English people. This difference is attributed partly to character of clothing and methods of heating, and partly to the fact that "the English people become *accustomed to cooler room temperature.*" This last statement indicates that the adjustments which lead to comfort have not been made in skin temperatures or the translations of these into sensations of heat or cold, but rather in the manner in which the sensations affect the individual. The assay of affective state, after all, is the objective of comfort studies. By training a person can gradually learn to expand or contract the zone within which he is comfortable. The climate is doubtless an important factor in this training and the adjustment is then attributed to acclimatization. It would seem that the separation of the summer comfort zone into climatic division is warranted on the basis of our present knowledge, and the committee is to be congratulated on this very practical suggestion. It is hoped that with a clearer understanding of the underlying factors the individual's selection of comfortable conditions can be predicted with a high degree of accuracy.

JOHN HOWATT, Chicago (WRITTEN): This paper is admittedly a summarization of research reports and other papers of pertinent interest on the general subject of comfort with summer air conditioning, and so makes little attempt to introduce any new or startling idea to the knowledge on the subject. It is an easily readable account of the progress and findings in the field and because of its readability will bring the facts that had been developed to the consciousness of many new readers.

The suggestion that the United States be divided into four zones from north to south in which effective temperatures of 70, 71, 72 and 73 deg are proposed for optimum comfort in summer is perhaps the only controversial idea introduced in the paper. It is admitted that climatization is a factor in determining optimum atmospheric conditions for comfort so it must be admitted that recognition of this factor must prevail; it must be admitted too that even when allowance has been made for all controllable factors, the field of air conditioning for summer comfort will always be one fruitful in criticism and complaint because the criticism and complaints come from individuals and we cannot hope to air condition for the optimum comfort of each individual. For example, how can room temperatures be maintained that will be satisfactory to a woman wearing six ounces of clothing and a man wearing six pounds, or for persons in the vital, vigorous age bracket of the teen years in the same room with people in the late years of life where fires are already dying out.

The use of effective temperature as a measure in the establishment of optimum comfort conditions in air conditioned space is perhaps as good as any that has been devised; nevertheless there are now many people becoming critical of using such a standard. Recent research presented to our Society through papers has tended to show the little effect relative humidity has in individual comfort provided the range is kept somewhere between 20 and 65 per cent and relative humidity is one of the factors in the effective temperature scale.

[9] The Summer Comfort Zone: Climate and Clothing, C. P. Yaglou and Philip Drinker. (ASHVE TRANSACTIONS, Vol. 35, 1929.)
[10] Loc. Cit. see Note 7.

On the whole this paper is a worthwhile contribution to our literature on the subject of summer air conditioning and is written in the style which I personally hope will be used more and more by writers on technical subjects so the papers themselves will be more readable.

R. R. SAYERS,[11] Washington, D. C. (WRITTEN): As air conditioning in Washington, D. C., is mentioned in this report it might be of interest to give some of the reactions of Government employees to air conditioning. Apparently it is the concensus of opinion that air conditioning is by no means perfect or entirely satisfactory, although persons who have been exposed to it would not want to give it up, especially in the summer. Persons who are acclimated to conditions that prevail in summer in Washington, D. C., however, suffer considerable discomfort from the daily change from a cool, northern to a hot, southern climate. A summer temperature of 80 F or 85 F during the day is not unpleasant for persons who are acclimated and doing sedentary work as are most of those occupying air-conditioned buildings in Washington.

When the employees of the Interior Department first moved into the new air-conditioned building the temperature was kept at about 70 F during the hottest weather. There was much complaint and many occupants tried various devices for shutting off the air and surreptitiously opened the windows until they were informed that anyone who opened a window would be subject to punitive measures. It developed that the contractor had not yet formally turned the building over to the Department and as one of the terms of the contract was that during a certain period the air in the building must be kept at 70 F regardless of outside air conditions, the employees had to submit to the discomfort until the contract was accepted by the Government. (Later, of course, the great difference between the inside and outside atmosphere was reduced somewhat.) As a consequence during the first summer the entering and leaving shock was most disagreeable and probably dangerous in some cases (at least one case of pneumonia developed). In fact the shock was so great to some that they remained in the building until after sundown to prevent possible collapse. The entrance shock was disagreeable but not so dangerous as the leaving shock. Women said that when they entered the building they felt as though they had stepped suddenly into cold water up to their knees.

In the summer, with a climate like that of Washington, D. C., it would not be practicable, to say the least, to wear warm clothing indoors as the heat and humidity out of doors sometimes are so great that enough clothing to make the contrastingly cold inside temperature comfortable could not be borne outside and clothing would have to be changed after entering and before leaving the air-conditioned atmosphere. To promote comfort and to prevent too great shock upon entering and leaving the inside temperature should be around 80 F to 85 F when the outside temperature is around 90 F to 95 F. The temperatures mentioned in the paper may be satisfactory for the Senate Chamber but it should be remembered that the members probably are exercising more or less (debating and moving about) while in most of the departmental offices the occupants are sitting still all day at desks. It is surprising what difference a little exercise makes. For instance, typists are comfortable at a lower temperature than other persons in the same room who are not so actively employed. Following are some of the complaints:

1. Never suffered from the heat so much at night as after spending 7 or 8 hours a day in air-conditioned buildings.
2. Too much draft at the windows. Occupants of buildings have to sit near the windows for light and the constant streams of air blowing on the back or sides of their bodies is very disagreeable and painful to those with a tendency toward rheumatism or arthritis. Some complain of constant stiff neck. Constant exposure of the body to moving air, even hot air, is annoying to say the least. This was one of the great disadvantages of the old system of window ventilation—those near the windows wanted them closed and those away wanted them open.
3. Too much difference between the inside air-conditioned atmosphere and the outside.
4. Sinus or throat trouble is not helped by exposure to air conditioning (some doctors in Washington, state that 9 of 10 persons in the District of Columbia are suffering with such troubles).

[11] Director, Bureau of Mines, Department of the Interior.

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5. Apparently there is no foundation for the claim that air conditioning prevents colds; once caught they are harder to cure. Doctors in the District of Columbia have been quoted as saying that, *This air conditioning is making millionaires of us.* Others say that their nurses complain that no vacations can be given since there is now no slack period during the summer such as before the introduction of air conditioning in the office buildings.

6. The air in air-conditioned rooms is not fresh and invigorating, however cool it may be and many persons complain of getting drowsy. The fact that this condition has been remedied by opening a window and getting some fresh air indicates that something is wrong with the inside air. The occupants of some of these air-conditioned buildings wonder how often the same air is recirculated.

It is possible that if the air conditioning was operated as recommended by the manufacturers of equipment it would be more satisfactory. However, to save expenses the air probably is recirculated too often.

There are delightful, invigorating days in the spring and fall when it would be much pleasanter, and perhaps healthier, to admit the sun-aerated air instead of the recirculated, more or less denatured, conditioned air. If air conditioning could be turned on and off like automatic heat it might be more satisfactory.

The impossibility of pleasing everyone is evident in the air-conditioned buildings. That was the principal objection to the old method of window ventilation. Some occupants of air-conditioned rooms want the air turned off and others want it on. If the drafts could be eliminated there would be a reduction in such complaints. Some persons have to sit too close to the windows to allow other occupants to get close enough to the light. If fresh air were supplied continuously in the winter, air conditioning during that season would be satisfactory as the draft is not so disagreeable when the air is warm. That the air is not fresh is indicated by the involuntary exclamation *how good the fresh air feels,* heard frequently when the occupants step out of an air-conditioned building.

To sum up the complaints against air conditioning, it is more or less uncomfortable and even detrimental to the health of some who are not in good physical condition and unfortunately many persons doing sedentary work are below par in this respect.

In spite of the complaints, very few if any employees in Washington who have become accustomed to air conditioning would want to forego it in summer, as more and better work can be done and street noises and dust do not readily penetrate closed windows. However, the question has been raised as to the possible effect on health of continual exposure to a climate already thought by some to be too stimulating. The "letting down" in activity during the less stimulating summer months may be a safety valve to prevent too rapid wearing out of the human machine.

The objections to air conditioning are mentioned as possible indications for future research toward perfecting a method of providing an atmospheric milieu having great potentialities for fostering human health and comfort.

C.-E. A. WINSLOW, New Haven, Conn.: The comfort votes reported in this paper are entirely in accord with physiological studies. For instance, we find that at the lower end of the hot zone, sweat secretion sets in more readily in summer than in winter, in other words, persons have been seasonly acclimated to that situation. At the upper exposure and particularly in dry climates there is an enormous range of adaptation. The ordinary individual in a hot, desert climate, can produce about a pint of water an hour and after exercise in such climates this can be raised to double or treble that amount. These are the phenomena that are reflected in the varying comfort zones.

C. E. BENTLEY, San Francisco, Calif.: We find in areas where it is extremely warm if low effective temperatures are maintained a shock effect takes place which is a point I believe should be taken into consideration.

C. TASKER, Toronto, Ont., Canada: I believe Mr. Chester should be extended a vote of thanks for presenting a paper of this form for it summarizes the work done by the Committee in a very readable form. With reference to Mr. Bentley's comment I would like to state that in all of the studies made, when I was the chairman of the Research Technical Advisory Committee of Sensations of Comfort, we very definitely took the most careful data of the outside weather conditions, because in

many of these cases, tests were made by subjects who entered the air conditioning enclosure from the outside.

These reactions were taken immediately upon entering and at frequent intervals for the first half hour and at less frequent intervals for the remainder of the test. Then data were collected again when the individuals left the air conditioned space and their reactions were taken immediately thereafter at regular intervals for a short time after leaving the enclosure.

Therefore, we did collect a lot of information pertaining to entering and leaving shock, but I think all of us realize that this is a wide subject on which there is a great need for more work. I agree with Mr. Chester that we can only hope to set up a band, and I believe that those of us who have been in the work would never dare to suggest that Minneapolis, Washington or San Francisco should always hold a certain effective temperature.

MR. CHESTER: Some time ago our Committee suggested that the Committee on Research provide funds for investigating the question of shock, as this is a question which the air conditioning engineer has to seriously consider. The public looks to the engineer to provide comfort and if in providing comfort an unfortunate condition results in subjecting people to serious shock, that is something we have to recognize and try to alleviate. The public in leaving a summer air conditioned space go out into the outside atmosphere where they feel as if they are entering an oven. This criticism should not be our most important consideration, because this is merely a recognition of the comfort the engineer has provided. Obviously it is not within our province to attempt to cool off the outside atmosphere, so in my opinion this should be disregarded and we should attempt to alleviate the undesirable shock upon entering an enclosure. In the past it has been suggested that anterooms could be provided, as they are in some places in Washington in the government buildings, which constitute intermediate zones so that individuals are not subjected to immediate shock, either upon entering or leaving.

With reference to the question by Dr. Keeton about people preferring a higher temperature in this country during the winter than in England, it should be mentioned that this variation is probably due to the difference in wall temperatures maintained in the two countries. For instance, in Chicago, Detroit or New York, the temperature drops to 0 and if you provide 70 F dry-bulb during the winter the inside wall surface temperature is around 40 F. In England where it is 40 F outside and it is common to maintain around 62 F dry-bulb temperature, the interior wall surface temperature is a good deal warmer because the outside wall surfaces seldom get below 40 F. Therefore, the loss by radiation from the skin to the wall is quite different and that together with the difference in clothing worn explains why English people have different conditions for comfort in winter.

F. W. HUTCHINSON, Berkeley, Calif.: I would like to add one item which may be of interest in substantiating the report of this Committee. Last summer I had occasion to travel by boat to Australia with 600 passengers, most of whom were Americans. We stopped at the Fiji Islands where 400 more British colonials were picked up, who were acclimated to the conditions at Fiji. From the Fiji Islands to Australia it was almost impossible to satisfy the people from Australia and the people from Fiji. The dining room was unbearable at all times, due to the fact that it was either too cold or too hot which is a fact believed to be of interest in connection with this paper. Charles Darwin, in one of his earlier books, mentioned the fact that in passing around Tierra del Fuego he found that when camp-fires were built the English huddled around the fire, whereas the natives, at a distance of 20 ft, were sweating profusely because they were acclimated.

MR. CHESTER: The last remark reminds me of the American who went to Patagonia some time ago. It was not very cold there, but there was a high wind. He was talking to a native who did not have much clothing and he inquired how he could stand the climate. The Patagonian said, "Americans don't wear clothing on face. Patagonian all face."

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The information given by Dr. Keeton very definitely supports the statement made at the beginning of the paper, that comfort is promoted when the air between the skin and clothing is maintained in a sub-tropical condition. He also supports the statement that differences in the comfort reactions of men and women can be attributed to differences in their clothing. His remarks and the references he cites form a very useful addition to the paper.

Mr. Howatt emphasizes the effects of disparities in clothing and also of differences in metabolic heat of individuals in widely varied age groups. Although he is correct in stating that relative humidity has little effect on comfort within a range of 25 to 65 per cent, it must not be forgotten that the extremes of this range involve a difference of 5 F in dry-bulb temperature at 71 deg ET. The effect of variations of relative humidity within the range he mentions, are not perceptible if they are along the same Effective Temperature line with properly correlated changes in dry-bulb temperature. However, with a constant dry-bulb temperature of say 76 F a variation of 40 per cent in relative humidity would have a very pronounced effect on comfort; in fact the average person would notice a change of 10 per cent in relative humidity if quickly made. Mr. Howatt is mistaken in surmising that the paper merely summarizes the research reports and other papers of pertinent interest and therefore introduces little new material. Actually most of the paper has not been taken from the sources mentioned but on the contrary represents the opinions of the authors.

Dr. Sayers' remarks are of considerable interest as they disclose the opinions of those who benefit, if this expression can be used in the face of some of his statements, from the operation of air conditioning systems. It would seem, however, that in reading the paper he has misread Effective Temperatures as dry-bulb temperatures. The table on atmospheric conditions in the Senate Chamber states that 72.7 deg ET is maintained when the outdoor air is between 91 and 95 F. It is subsequently stated that the relative humidity indoors was 50 per cent and this means that the dry-bulb temperature within the Senate Chamber was 78 F which is considerably warmer than the temperature of 70 F in the Department of the Interior Building, which he states gave rise to many complaints. He is right in calling attention to the desirability of supplying more fresh air to air conditioned buildings, but of course this would increase the first cost and the operating expense and whenever such important matters can be disregarded, which is seldom, ideal indoor atmospheric conditions can be maintained. He also makes a good point in stating that ventilation should be increased when this can be done at little expense, which is in the spring and fall when little and sometimes no change is required from the condition of the outdoor air. Unfortunately in most cases air distributing and return systems and air filters are designed on the basis of air volumes required with full use of the conditioning means and complete or partial by-passing of the cooling coils or spray type dehumidifiers would not permit much increase in volumetric capacity, due to the fact that the resistance to flow increases as the square of the velocity. It perhaps would be worth while to resort to two speed fan drives for the purpose of increasing the intake of fresh air under suitable weather conditions, as there are relatively cool periods even in summer. In many instances existing systems could be improved by the addition of auxiliary minimum fresh air supply dampers under the control of outdoor air thermostats. Also operating engineers can increase the intake of fresh air when desirable with most of the existing plants up to the capacities of the fresh air inlets.

Dr. Winslow's remarks constitute an authoritative endorsement of the recommended division of the United States into four different Effective Temperature zones.

Mr. Tasker is correct in pointing out that unvarying indoor Effective Temperatures are not advocated. They should of course be increased as summer advances and be decreased as the heat abates. The Effective Temperatures given on the map are for hot days in July and as stated in the paper they can be modified in accordance with such important factors as building construction, solar exposure, density of occupancy, degree of activity of the occupants, and last but not least the money available for the purchase of air conditioning equipment.

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No. 1197

USE OF COLD ACCUMULATORS IN THE AIR CONDITIONING FIELD

By R. W. Evans* and C. J. Otterholm,** Minneapolis, Minn.

Section I—Introduction

THE cold accumulator was first developed to store *refrigeration effect* to meet the short period refrigeration demands, found in the Dairy Industry. Similar needs for a refrigeration *storage battery* are found in many types of air conditioning work.

Considerable data are now available for the determination of peak cooling demand. These, however, are insufficient for the selection of accumulator equipment, since such selection is dependent upon the total cooling demand in a 24-hour period.

This paper concerns the introduction of new factors, such as, *Accumulated Degree-Hours* and *Accumulated Sun Intensity Values* to be applied in the accurate determination of total cooling load. The performance characteristics of one type of cold accumulator and its application to air conditioning loads of several types is also discussed, together with certain specific economies obtainable with this method of cooling.

Section II—Performance Characteristics of a Typical Cold Accumulator

The type of accumulator discussed in this paper originated as a plan to utilize water as a substitute for brine in dairy and creamery plants. A very satisfactory unit was developed and based on the principle of using thin sheets of ice, formed on copper plates, with high rates of water circulation. It was found possible to cool milk and cream to 40 F just as quickly with 35 F water plus high velocity circulation as was ordinarily obtained with 20 F brine. Over 200 of these cold accumulators are now in daily operation in the United States and Canada.

The air conditioning accumulator presented no new problems in design, except those relating to increased storage capacity. Due to the wide range of air conditioning loads, requiring corresponding wide ranges of storage capacity, it was decided to standardize on one size unit, having a nominal storage capacity of 24 ton-hours of cooling effect, and installing as many

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units or sections, as may be required to meet any air conditioning load requirement. The manufacture of but one size unit made possible the securing of extremely accurate performance data, simple and flexible installation arrangements and lowest possible manufacturing cost.

Fig. 1 is a cross section, illustrating the interior construction. In general the accumulator consists of a steel tank and a series of copper plate coils, arranged to form a long labyrinthian passage way for the circulating water stream. During the accumulating or charging period, thin sheets of ice are formed on each side of the plate coils under automatic control. Due to the

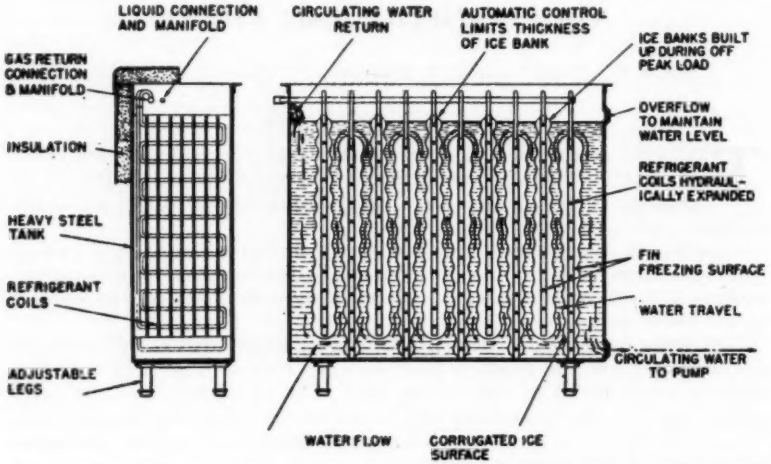

FIG. 1. TYPICAL ILLUSTRATION OF THE GENERAL DESIGN OF ACCUMULATOR FOR AIR CONDITIONING APPLICATIONS

high latent heat of fusion of water in forming ice, a large amount of *refrigeration effect* is stored compactly. During the utilization or discharge period, the ice is melted by a stream of water circulating between the air conditioning coil or unit and the accumulator.

In order to control the thickness of the ice formation on each plate or baffle the refrigerant is fed independently and in progressively varying amounts to each element. Likewise the suction connection from each plate is connected to a suction header in such manner as to provide varying suction pressure (and temperature) for each element and in a counter-progressive sequence, as related to the liquid refrigerant supply.

PERFORMANCE TESTS

(a) *Apparatus and Method:* A single fully charged accumulator was connected through a water circulating line to a gas-fired water heater. The temperature of the circulating water was controlled to maintain a constant temperature difference of 6 F between the inlet to and outlet from the accumulator. A large number of complete tests was conducted, for different water

circulating rates; the water circulation was measured and maintained at a constant value for each test.

Because the water circulation and the difference between the inlet and outlet water temperatures were maintained at constant values for each test, the accumulator was discharged at a constant rate in Btu per minute.

	WATER CIRCULATING RATE GPM	DISCHARGE RATE BTU PER MIN
1	15	750
2	22.5	1125
3	30	1500
4	37.5	1875
5	45	2250
6	60	3000

The temperature of the water leaving the accumulator was measured and tabulated and the elapsed time noted for each 1 F increase of leaving water temperature.

The results of these tests were plotted for several values of continuous water flow, six of which are shown in Fig. 2.

(b) *Analysis of Test Data:* The tests show that the available cooling effect or capacity which can be obtained from a single accumulator depends upon the temperature of the cooling water required.

In air conditioning applications, cooling water temperatures of from 40 F to 50 F are required, the exact requirement being determined by the latent to total heat gain ratio.

The total available cooling effect which can be obtained from one of these particular accumulators may be determined by selecting the desired cooling water temperature, following the horizontal temperature line to the right in Fig. 2 to the intersection of the desired water circulating rate curve obtained by calculation from the peak hour heat gain, and multiplying the corresponding accumulator discharge rate in Btu per minute at this intersection by the elapsed time in minutes. This product is the net available Btu capacity of the accumulator when delivering this temperature water.

For example, assume the following:

Calculated peak hour heat gain 90,000 Btu per hour
Latent to total heat gain ratio such that 40 F cooling water is required

$$\frac{\text{90,000 Btu per hour}}{\text{8.33 lb per gal} \times \text{60 min} \times \text{6 deg temp rise circulating rate}} = \text{30 gpm water}$$

The tests described were based upon an allowable water temperature rise of 6 F. Other performance curves are available based upon tests for other allowable temperature rises.

From Fig. 2 follow 40 F horizontal water temperature line to intersection with 30 gpm water circulating curve, equivalent to a discharge rate of 1500 Btu per minute, and read 206 min of elapsed time for water delivered by the accumulator *to reach this final outlet temperature.*

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1500 Btu per minute × 206 min = 309,000 Btu accumulator capacity available or 25¾ ton-hours

Assume this same calculated peak hour heat gain and latent to total heat gain ratio but with a calculated total cooling load of 50 ton-hours, (by methods hereinafter described).

A total water circulating rate of 30 gpm is still required. Two accumulators are indicated since the capacity of one accumulator when delivering 40 F

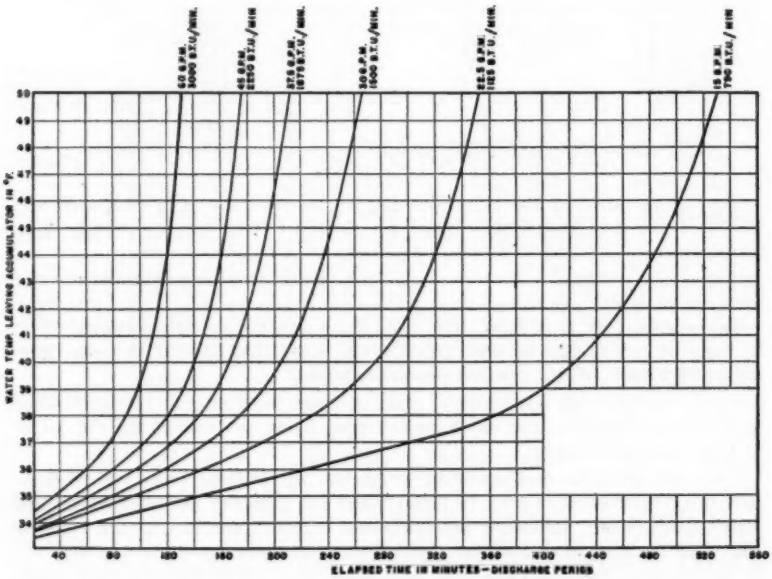

FIG. 2. PERFORMANCE CURVES FOR AIR CONDITIONING APPLICATIONS BASED ON 6 DEG DROP IN WATER TEMPERATURE THROUGH ACCUMULATOR

cooling water is 25¾ ton-hours. The water circulating rate through each of the two accumulators would be ½ of 30 gpm or 15 gpm.

Following the 40 F water temperature line to the intersection with the 15 gpm water circulating curve, equivalent to a discharge rate of 750 Btu per minute, and read 424 min of elapsed time for water delivered by each accumulator to reach this final outlet temperature.

750 Btu per minute × 424 min = 318,000 Btu accumulator capacity of each unit
or 26½ ton-hours
2 × 26½ ton-hours = 53 ton-hours total accumulator capacity (2 Units)

In like fashion any combination of peak hour demand and total ton-hours of cooling load can be met by proper choice of circulating rate and accumulator units.

Analysis of test data shows that the total ton-hours output of one such accumulator is dependent only upon the temperature of the water to be delivered by the accumulator, regardless of the total water circulation required. The rate of water circulation through the accumulator determines the rate at which its capacity is to be utilized.

Using Fig. 2 and assuming various accumulator capacity values (output values) and dividing these assumed capacity values, expressed in Btu, by the discharge rate, in Btu per minute, corresponding to various water circulation rates, the result will be expressed in elapsed time, in minutes, required for the accumulator to deliver the assumed capacities. The intersection of these elapsed time vertical lines with the water circulation curve will show the

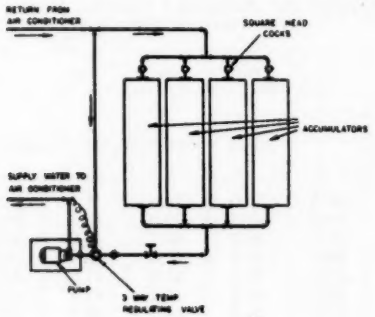

FIG. 3. TYPICAL PIPING DIAGRAM OF A MULTIPLE ACCUMULATOR INSTALLATION

terminal temperatures of the cooling water being delivered by the accumulator while discharging these assumed capacities.

Table 1 has been constructed on this basis and capacity values given are for one accumulator. For multiple installation it is only necessary to multiply these capacities or output values by the number of accumulators used.

If 50 F cooling water is required approximately 33 ton-hours of cooling effect is available from a single unit, whereas if 40 F cooling water is required, approximately 26 ton-hours of cooling effect would be available from a single unit.

The 33 ton-hours could be delivered in 2 hours, 12 min if 60 gpm of water circulation is used through the single unit or extended to a delivery period of 8 hours, 50 min if 15 gpm of water circulation is used.

The 26 ton-hours could be delivered in 1 hour, 44 min, if 60 gpm of water circulation is used or extended to a delivery period of 7 hours, 4 min if 15 gpm of water circulation is used.

Large cooling loads are easily handled by selecting the correct number of accumulators to provide the total storage capacity required. For example,

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if an application indicated the need of a total storage capacity of 300 ton-hours, with circulating cooling water of 44 F, 10 accumulators should be installed.

If the peak hour heat gain calculation indicated that the highest demand on the system would be 900,000 Btu per hour (15,000 Btu per minute) it would be necessary to circulate

$$\frac{15{,}000}{8.33 \times 6 \text{ deg}} = 300$$

gpm, or 30 gpm per accumulator.

Charging tests have shown that a compressor operates at an average suction refrigerant temperature of 24 F when its capacity is 10 per cent or

TABLE 1—CAPACITY AND AVAILABLE COOLING WATER TEMPERATURE ONE UNIT ACCUMULATOR

FLOW GPM		15	22.5	30	37.5	45	60	
OUTPUT		FINAL WATER TEMP DEG FAHRENHEIT						AVG
BTU	TON-HOURS							
288,000	24	38.5	39.0	39.0	38.7	38.6	38.7	38.8
300,000	25	39.0	39.5	39.6	39.3	39.1	39.3	39.3
312,000	26	39.7	40.1	40.2	40.0	39.8	40.0	40.0
324,000	27	40.4	40.9	41.0	40.9	40.5	40.8	40.8
336,000	28	41.3	41.8	42.0	41.8	41.4	41.6	41.7
348,000	29	42.4	42.8	43.2	42.8	42.5	42.6	42.7
360,000	30	43.7	44.1	44.5	44.0	43.9	43.8	44.0
372,000	31	45.3	45.9	46.0	45.7	45.5	45.5	45.7
384,000	32	47.3	47.7	47.5	47.5	47.3	47.8	47.5
396,000	33	49.7	49.9	49.6	49.8	49.9	50.0	49.8

less of the total accumulator capacity. The size of the charging compressor to be used will depend upon the hours of charging time available, before the next discharge period begins. Approximately 10,000 Btu per hour per horsepower can be delivered by the conventional water cooled compressor unit at 24 F suction temperature.

Fig. 3 shows a typical piping diagram of a multiple accumulator installation. A bypass line is installed between the main supply and return water lines to and from the air conditioner through a 3-way temperature control valve, the control element of which is attached to the cold water supply line to the air conditioner. This control element is set to maintain a constant supply water temperature delivered to the air conditioner. Square head cocks are installed in the return water line to each accumulator to balance the water flow through each unit. The circulating pump is only operated during the discharge period, no water circulation being required during the charging period.

SECTION III—CALCULATION OF TOTAL COOLING LOAD

To apply a cold accumulator properly, it is necessary to determine the total accumulated cooling load for a given cooling period.

The various components of cooling load must be determined separately in any heat gain calculation to obtain either hourly peak demand or total operating period load.

In the determination of *hourly peak heat gain* from conduction through walls, windows, roofs, etc., multiply these various areas by unit conduction coefficients and design temperature differences between the inside and outside of the enclosure to be cooled.

The accumulated conduction cooling load would similarly be obtained if the product of these same areas and conduction coefficients were multiplied by previously determined *accumulated degree-hours* for a given cooling period. The same accumulated *degree hours* would be used to calculate the total sensible heat load from ventilation air introduced into the cooled space. In the same manner, the total cooling load due to sun effect would be determined

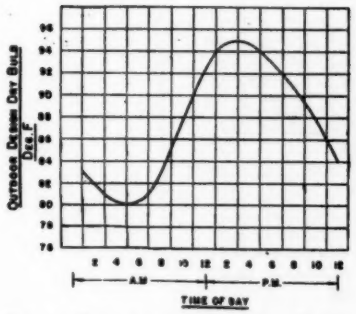

FIG. 4. REPRODUCTION OF CURVE SHOWING TYPICAL VARIATIONS OF OUTDOOR TEMPERATURES FOR 24-HOUR PERIOD

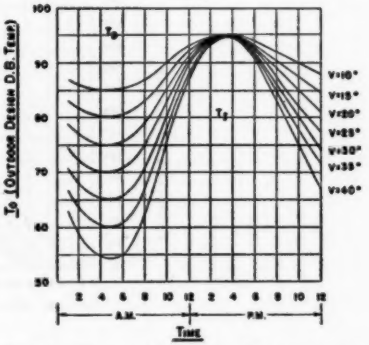

FIG. 5. TYPICAL OUTDOOR TEMPERATURE VARIATION CURVE FOR SEVEN VALUES OF JULY MEAN TEMPERATURE RANGES

for any given operating period if the product of exposed wall, roof or window areas and reflection and conduction factors were multiplied by previously determined *accumulated sun intensity* values.

(A) *Tables of Accumulated Degree Hours:* Fig. 4 is a reproduction[1] of a curve showing typical variations of outdoor temperatures for a 24-hour period.

This temperature variation curve is based upon the mean difference between the daily maximum and daily minimum outdoor temperatures for the month of July, as explained in the text of this paper.[2] This mean difference is termed by the U. S. Weather Bureau as the *mean daily range* for July.

In the ASHVE GUIDE, 1941, p. 126, is tabulated outdoor *design* values of dry-bulb and wet-bulb temperatures recommended for heat gain calculations

[1] A Rational Heat Gain Method for the Determination of Air Conditioning Cooling Loads, by F. H. Faust, L. Levine and F. O. Urban. (ASHVE TRANSACTIONS, Vol. 41, 1935.)

[2] Effect of Heat Storage and Variation in Outdoor Temperature and Solar Intensity on Heat Transfer Through Walls, by J. S. Alford, J. E. Ryan and F. O. Urban. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

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for various points in the United States. Table 2 reproduces these design dry-bulb and wet-bulb values and adds a third value entitled *design mean temperature range* for these same communities.

The design mean temperature range values, given in the table, have been determined from an examination of 1940 weather reports and comparison of these values with the normal July mean temperature range given in most weather reports.

While the values given in the table are believed to be accurate, it would be advisable, before selecting a design mean temperature range, for accu-

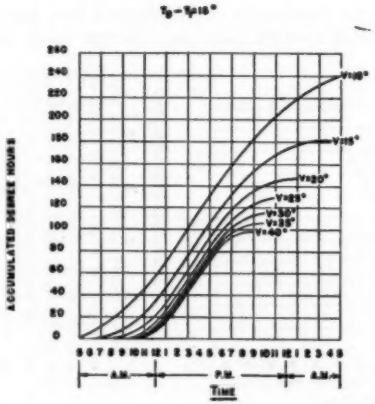

FIG. 6. TYPICAL ACCUMULATED DEGREE-HOUR CURVE FOR A DESIGN MEAN TEMPERATURE RANGE OF 15 F

mulated load calculations hereinafter described, to check with the nearest weather bureau to obtain the normal July mean temperature range for that particular locality.

Fig. 5 is a typical outdoor temperature variation curve for seven values of July mean temperature ranges (V = 10 F, 15 F, 20 F, 25 F, 30 F, 35 F, 40 F). These curves are based upon an outdoor design dry-bulb temperature of 95 F. An indoor design dry-bulb temperature of 80 F is also indicated in Fig. 5, to show the effect of a small or large mean temperature range on the total 24-hour cooling requirement.

Reference to Table 2 shows that most design values of July mean temperature range fall between 10 F and 20 F.

In Tables 3-A, B and C, are given *accumulated degree hours* for five design values of mean temperature range (10 F, 12.5 F, 15 F, 17.5 F, 20 F). These accumulated degree hours were obtained by plotting typical outdoor tempera-

ture variation curves similar to Fig. 5 and integrating the area of the curves above assumed values of indoor design temperatures. Table 3-A is to be used for a design indoor-outdoor dry-bulb temperature difference of 12 F; Table 3-B, for a design indoor-outdoor dry-bulb temperature difference of 15 F; Table 3-C, for a design indoor-outdoor dry-bulb temperature difference of 18 F.

In general, the 12 F indoor-outdoor design temperature difference will be used where the maximum outdoor design dry-bulb temperature is from 88 F

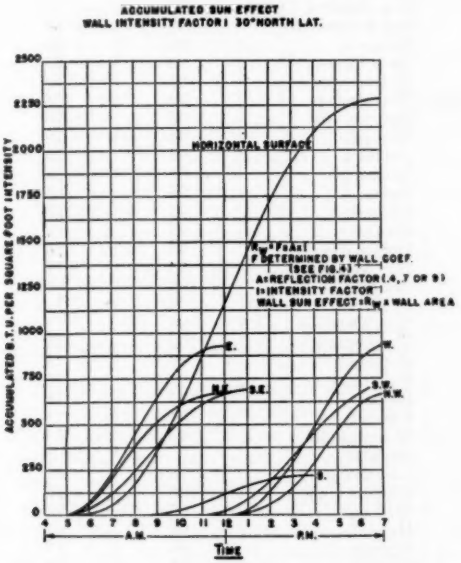

FIG. 7. GRAPHIC ILLUSTRATION OF THE ACCUMULATED ACTION OF THE SUN SHINING ON VARIOUS EXPOSED SURFACES THROUGHOUT TYPICAL JULY DAYS—30 DEG NORTH LATITUDE

to 92 F, the 15 F value where the maximum outdoor design dry-bulb temperature is from 93 F to 97 F and the 18 F value where the maximum outdoor design dry-bulb temperature is from 98 F to 100 F.

In using these tables, the *difference* in degree hours between the start and finish of a cooling period is to be used. For example, assume a theater operating from noon until midnight (12 hours) in a community where the outdoor design dry-bulb temperature, T_o, is 92 F; indoor design dry-bulb temperature, T_1, is 80 F; and design mean temperature range, V, is 15 F. From Table 3-A, Column 3 ($V = 15$ F).

Accumulated degree hours at midnight	121
Accumulated degree hours at noon	18
Design degree-hours	103

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TABLE 2—DESIGN OUTDOOR TEMPERATURES

STATE	CITY	DESIGN DRY-BULB	DESIGN WET-BULB	DESIGN MEAN TEMPERATURE RANGE
Alabama	Birmingham	95	78	15
	Mobile	95	80	15
Arizona	Phoenix	105	76	20
Arkansas	Little Rock	95	78	15
California	Los Angeles	90	70	20
	San Francisco	90	65	10
Colorado	Denver	95	64	22.5
Connecticut	New Haven	95	75	20
Delaware	Wilmington	95	78	20
District of Columbia	Washington	95	78	15
Florida	Jacksonville	95	78	15
	Tampa	94	79	12.5
Georgia	Atlanta	95	76	15
	Savannah	95	78	15
Idaho	Boise	95	65	25
Illinois	Chicago	95	75	15
	Peoria	95	76	20
Indiana	Indianapolis	95	76	20
Iowa	Des Moines	95	77	20
Kansas	Wichita	100	75	15
Kentucky	Louisville	95	76	15
Louisiana	New Orleans	95	79	12.5
	Shreveport	100	78	15
Maine	Portland	90	73	15
Maryland	Baltimore	95	78	15
Massachusetts	Boston	92	75	17.5
Michigan	Detroit	95	75	17.5
Minnesota	Minneapolis	95	75	20
	Duluth	...	...	20
Mississippi	Vicksburg	95	78	15
Missouri	Kansas City	100	76	20
	St. Louis	95	78	17.5
Montana	Helena	95	67	25
Nebraska	Lincoln	95	75	17.5
	Omaha	95	75	17.5
Nevada	Reno	95	65	30
New Hampshire	Concord	90	73	20
New Jersey	Trenton	95	78	20
New York	Albany	92	75	20
	Buffalo	93	75	15
	New York	95	75	20
New Mexico	Santa Fe	90	65	15
North Carolina	Asheville	90	75	15
	Wilmington	95	79	15
North Dakota	Bismarck	95	73	25
Ohio	Cincinnati	95	78	15
	Cleveland	95	75	15
Oklahoma	Oklahoma City	101	76	20
Oregon	Portland	90	65	20
Pennsylvania	Philadelphia	95	78	15
	Pittsburgh	95	75	20
Rhode Island	Providence	93	75	20
South Carolina	Charleston	95	80	12.5
	Greenville	95	76	15
South Dakota	Sioux Falls	95	75	20

TABLE 2—DESIGN OUTDOOR TEMPERATURES—*Continued*

STATE	CITY	DESIGN DRY-BULB	DESIGN WET-BULB	DESIGN MEAN TEMPERATURE RANGE
Tennessee	Chattanooga	95	77	15
	Memphis	95	78	15
Texas	Dallas	100	78	15
	El Paso	100	69	20
	Galveston	95	80	10
	Houston	95	78	15
	San Antonio	100	78	20
Utah	Salt Lake City	92	63	25
Vermont	Burlington	90	73	20
Virginia	Norfolk	95	78	15
	Richmond	95	78	17.5
Washington	Seattle	85	65	17.5
	Spokane	90	65	17.5
West Virginia	Parkersburg	95	75	20
Wisconsin	Madison	95	75	20
	Milwaukee	95	75	20
Wyoming	Cheyenne	95	65	25

Fig. 6 is a typical accumulated degree-hour curve for a design outdoor-indoor temperature difference of 15 F and for seven values of design mean temperature range. This curve graphically illustrates the data given in Table 3-B together with additional values for other design mean temperature ranges.

(B) *Tables of Accumulated Sun Intensity Values:* In the previously mentioned paper [3] are given sun effect coefficients at various latitudes for walls, roofs, windows and skylights facing several directions. The ASHVE GUIDE also presents similar tables for solar radiation factors expressed in Btu per square foot per hour. As explained in the text of Mr. Faust's (and others) paper, the wall sun effect intensity factors in the tables are to be multiplied by, first a reflection factor (0.4, 0.7 or 0.9) depending upon whether the wall under consideration is light colored, medium colored, or a black surface and secondly, a factor related to the conduction heat transmission coefficient of the wall or roof under consideration.

The tables applying to glass surfaces, such as windows or skylights, are to be used directly, multiplying such factors by the window or skylight areas to obtain heat transmission to an enclosure due to sun effect. Shading factors are then applied to obtain the percentage of sun effect heat transmission to the enclosure through window or skylight areas.

In Tables 4-A, B, C and D, the tables from the ASHVE TRANSACTIONS,[4] have been used for north latitudes of 30 deg, 35 deg, 40 deg, and 45 deg to obtain multipliers for exposed wall and glass areas to obtain both instantaneous and accumulated heat gain due to sun effect.

The wall sun effect factors are based upon a wall conduction coefficient, U_w, of 0.34 and a reflection factor for a medium colored wall, 0.7. The roof

[3] Loc Cit. Note 1 (p. 327).
[4] Loc Cit. Note 1 (pp. 339, 340).

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TABLE 3-A—ACCUMULATED DEGREE HOURS MAXIMUM DESIGN DRY-BULB TEMPERATURE DIFFERENTIAL = 12 F

JULY MEAN TEMP. RANGE	10 DEG	12.5 DEG	15 DEG	17.5 DEG	20 DEG
5 A.M.	...	...	...	...	...
6 A.M.	2	...	...	...	...
7 A.M.	5	...	...	...	...
8 A.M.	8	2	...	...	...
9 A.M.	13	5	1	...	...
10 A.M.	19	10	4	3	1
11 A.M.	27	17	10	6	5
12 Noon	37	26	18	14	12
1 P.M.	47	37	28	23	21
2 P.M.	59	48	39	34	32
3 P.M.	71	60	51	46	44
4 P.M.	83	72	63	58	56
5 P.M.	95	84	75	70	68
6 P M.	106	95	86	81	78
7 P.M.	116	106	95	90	87
8 P.M.	126	115	104	97	94
9 P.M.	134	123	110	103	99
10 P.M.	142	130	115	107	102
11 P.M.	148	135	119	109	103
12 Mid.	154	139	121	110	...
1 A.M.	158	142	122	...	...
2 A.M.	162	143	...	...	...
3 A.M.	165	...	...	...	...
4 A.M.	167	...	...	...	...
5 A.M.	169	...	...	...	...

TABLE 3-B—ACCUMULATED DEGREE HOURS MAXIMUM DESIGN TEMPERATURE DIFFERENTIAL = 15 F

JULY MEAN TEMP. RANGE	10 DEG	12.5 DEG	15 DEG	17.5 DEG	20 DEG
5 A.M.	...	...	...	...	...
6 A.M.	5	...	...	...	...
7 A.M.	11	5	1	...	...
8 A.M.	17	9	3	1	...
9 A.M.	25	15	7	4	1
10 A.M.	34	24	13	9	5
11 A.M.	45	34	22	17	12
12 Noon	58	46	34	28	22
1 P.M.	71	64	46	40	34
2 P.M.	86	74	61	55	48
3 P.M.	101	88	76	70	63
4 P.M.	116	104	91	85	78
5 P.M.	131	118	105	99	92
6 P.M.	145	132	119	113	106
7 P.M.	158	145	132	125	118
8 P.M.	171	157	143	136	128
9 P.M.	182	168	153	145	136
10 P.M.	194	178	161	152	142
11 P.M.	203	186	168	157	146
12 Mid.	212	193	173	160	147
1 A.M.	219	198	177	162	...
2 A.M.	226	203	180	163	...
3 A.M.	232	207	181	...	...
4 A.M.	237	210	...	...	...
5 A.M.	242	...	...	...	...

TABLE 3-C—ACCUMULATED DEGREE HOURS MAXIMUM DESIGN TEMPERATURE DIFFERENTIAL = 18 F

JULY MEAN TEMP. RANGE	10 DEG	12.5 DEG	15 DEG	17.5 DEG	20 DEG
5 A.M.	...	...	...	...	...
6 A.M.	8	6	4	2	...
7 A.M.	17	13	8	4	...
8 A.M.	26	20	13	7	1
9 A.M.	37	29	20	13	5
10 A.M.	49	39	29	20	11
11 A.M.	63	52	41	31	21
12 Noon	79	67	55	45	34
1 P.M.	95	83	71	60	49
2 P.M.	113	101	88	77	66
3 P.M.	131	129	106	95	84
4 P.M.	149	137	124	113	102
5 P.M.	166	152	137	129	120
6 P.M.	184	169	154	145	136
7 P.M.	200	185	169	160	151
8 P.M.	216	200	184	174	164
9 P.M.	230	213	196	186	175
10 P.M.	244	226	208	196	184
11 P.M.	256	237	217	204	191
12 Mid.	268	247	226	211	196
1 A.M.	278	255	232	216	199
2 A.M.	288	263	237	219	200
3 A.M.	296	269	242	221	...
4 A.M.	305	276	246	223	...
5 A.M.	313	281	249	...	...

sun effect coefficients are based upon a roof conduction coefficient, U_r, of 0.2 and a reflection factor of a black surface, 0.9. Correction factors for other conduction coefficients and reflection factors are given in the tables.

The instantaneous values, for use in calculating peak heat gain are given in one column, and accumulated values in another column. The accumulated values were obtained by plotting instantaneous values and integrating the area under these curves.

In using the tables to obtain the accumulated load due to sun effect, the difference in values between the start and finish of a cooling period is to be

EXPOSURE	EAST	SOUTH	WEST	ROOF
Wall and Roof Factors				
Accumulated *Rw* (or *Rr*) Noon	52.2	6.4	None	48.5
Accumulated *Rw* (or *Rr*) 10:30 A.M. (By Interpolation)	47.3	2.1	None	30.6
Design *Rw* (or *Rr*)	4.9	4.3	None	17.9
Window Factor				
Accumulated *Rg* Noon	810	40	None	1020
Accumulated *Rg* 10:30 A.M. (By Interpolation)	752	12	None	630
Design *Rg*	58	28	None	390

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used. For example, assume a church with a morning service beginning at 10:30 A.M. and completed at 12:00 noon, the location of the church being 30 deg north latitude and with East, South, West and Roof exposures. Select accumulated sun effect factors as shown in tabulation on p. 135.

It should be borne in mind that in calculating the *peak hour* heat gain from sun effect, only such exposures are considered which have the sun shining on them at the particular time the *peak load* hour is determined, whereas, all outdoor exposures must be considered in calculating the *accumulated total* heat gain.

Fig. 7 graphically illustrates the accumulated action of the sun shining on various exposed surfaces throughout typical July days in a location of 30 deg North Latitude.

No account is taken in the table values of the effect of heat storage in the walls. There is, of course, a definite time lag in the delivery of outdoor heat to an enclosure due to heat storage in the structure. It is essential that the design engineer take into account the effect of this stored heat when calculating the total accumulated heat gain. The value to be set up to precool the structure and to be added to the total load calculation is left to the judgment of the design engineer. In general, it will fall between two and three times the calculated peak hour conduction and sun effect load; however, type of construction, insulation used and exposure will cause a variation in the amount to be added to the net calculated load.

In this connection, it may be added that *negative conduction* and radiation will result in some cases, where the outdoor temperatures drop below indoor design values after the sun goes down. This is effectively shown in Fig. 5, particularly where design mean temperature ranges are high. Such negative loads could be calculated and deducted from the total previously determined accumulated cooling load.

In calculating the total sensible cooling period load from occupancy, light and appliances, and latent cooling period load from occupancy, ventilation air and appliances it may be assumed that these loads are constant throughout the cooling period under consideration. It is therefore only necessary to multiply the *peak hour* value of these components of heat gain by the hours of the cooling cycle, to obtain their contribution to the total cooling load.

Peak hour heat gains are calculated in Btu per hour and similarly, using the table values of accumulated degree-hours and accumulated sun intensity values, the total cooling load will be expressed in Btu. The peak hour heat gain may be expressed in tons of cooling demand and the total load in ton-hours by dividing the calculated results by 12,000.

(C) *Selection of Required Accumulator Capacity:*
Example: Industrial Cafeteria

Total cooling load — 960,000 Btu = 80 ton-hours.
Peak hour cooling demand — 288,000 Btu per hour = 24 tons.

Cooling period	5 hours
Non-operating period	19 hours
Total	24 hours

46 F cooling water required to meet latent to total heat ratio.

Minimum charging capacity

$$\frac{80 \text{ ton-hours}}{24 \text{ hours}} = 3.33 \text{ tons}$$

Approximate capacity compressors at 24 F suction temperature 75 F cooling water.

3 hp — 2.2 tons
5 hp — 4.4 tons

Select 5 hp compressor.

(a) *Assume Charging Compressor Operating During Cooling Period*

5 hours × 4.4 tons = 22 ton-hours available from compressor operation.

80 ton-hours — 22 ton-hours = 58 ton-hours of accumulator capacity required.

Referring to Fig. 8, two accumulators will have a capacity of 60 ton-hours when delivering 46 F cooling water (intersection of 60 ton-hour and 46 F water lines).

Following the 60 ton-hour line to the right to the 4.4 ton per hour charging line, 13.6 hours of charging time will be required out of 19 hours available, leaving 5.4 hours shut down period (intersection of 4.4 ton charging line with 13.6 interpolated hour line).

(b) *Assume Charging Compressor Not Operating During Cooling Period*

80 ton-hours accumulator capacity required.

Referring to Fig. 8, three accumulators will have a capacity of 90 ton-hours when delivering 46 F cooling water (intersection of 90 ton-hour and 46 F water lines).

Following the 80 ton-hour storage capacity line to the right, to the 4.4 ton per hour charging line, 18.2 hours of charging time will be required out of 19 hours available. Note that if direct expansion cooling were to be used a 24 ton compressor would have to be installed to meet the peak hour demand, even though this compressor could be operated at a relatively higher suction temperature.

With the accumulators, the load demand, both peak hour and total, can be met with a 4.4 ton compressor.

SECTION IV—CONCLUSIONS

While the cold accumulator is particularly adapted to the handling of heavy air conditioning loads of short duration, this study of actual degree-hours of typical cooling loads for a month such as July indicates that the wide peaks and valleys in a resulting curve made up of hourly transmission, solar heat gain, ventilation load as well as variations in occupancy, make it possible, by the use of a suitable accumulator unit, to reduce the size of the compressor equipment materially, with resulting economies in first cost, operating expense and upkeep.

Further, the use of these new refrigeration *storage batteries* make practical the development of *off-peak* power loads which will reduce the cost of electric power to the owners of air conditioned buildings while making summer cooling loads more desirable to the electric utility company.

Since periods of *peak demand* on most utility systems rarely extend over more than four hours, it is simple to arrange control circuits so that the entire recharging of the accumulator cycle takes place *off-peak*, and particularly so in the case of hotel banquet halls, churches, auditoriums, mortuaries, and noonday cafeterias. In addition many office buildings and retail store

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Table 4-A—30 deg North Latitude Wall and Roof Sun Effect Factor R_w and R_r*

Time	N. E.		East		S. E.		South		S. W.		West		N. W.		Roof	
	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM
6 a.m.	7.6	2.4	7.6	2.6	2.6	1.7	—	—	.....	.....	.....	.....	.....	.....	1.1	0.5
7 a.m.	9.3	11.8	10.8	13.9	0.3	0.7	—	—	.....	.....	.....	.....	.....	.....	3.4	2.6
8 a.m.	9.0	20.9	11.8	30.4	7.6	12.7	—	—	.....	.....	.....	.....	.....	.....	6.4	7.4
9 a.m.	7.4	26.6	9.9	35.5	7.5	21.6	0.6	0.2	.....	.....	.....	.....	.....	.....	8.9	13.1
10 a.m.	4.2	28.4	7.6	41.2	6.7	28.6	1.7	1.4	.....	.....	.....	.....	.....	.....	10.8	24.9
11 a.m.	1.1	27.0	4.9	39.1	4.8	34.5	2.8	3.7	—	—	.....	.....	.....	.....	12.1	36.2
12 M.	—	22.2	—	32.1	2.0	37.5	3.5	6.2	—	—	.....	.....	.....	.....	12.5	46.5
1 p.m.	.....	.....	.....	.....	.....	.....	3.5	9.3	4.5	4.8	2.0	1.1	—	—	12.1	57.7
2 p.m.	.....	.....	.....	.....	.....	.....	3.4	12.3	7.5	10.6	4.7	4.2	1.1	0.6	10.8	67.5
3 p.m.	.....	.....	.....	.....	.....	.....	1.7	11.2	9.6	15.9	7.6	7.3	4.2	2.4	10.2	72.0
4 p.m.	.....	.....	.....	.....	.....	.....	0.6	12.2	7.8	17.9	10.4	11.9	7.0	4.6	8.9	71.7
5 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	7.6	23.5	11.8	17.5	9.0	10.5	6.4	69.3
6 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	6.1	23.8	10.9	20.7	9.3	20.1	3.4	62.8
7 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	3.7	27.1	7.8	47.9	7.0	34.2	1.1	50.2
8 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	—	30.6	—	82.2	—	37.7	—	37.5

*R_w factors are for wall conduction U-value = 0.24
R_r factors are for roof conduction U-value = 0.3

For other *wall* U-values multiply R_w factors by $\frac{U}{0.24}$

For other *roof* U-values multiply R_r factors by $\frac{U}{0.3}$

R_w factors are given for medium colored walls
For light colored walls—multiply R_w factors by 0.572
For very dark walls—multiply R_w factors by 1.285

Table 4-B—35 deg North Latitude Wall and Roof Sun Effect Factor R_w and R_r*

Time	N. E.		East		S. E.		South		S. W.		West		N. W.		Roof	
	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM
6 a.m.	7.0	3.9	8.1	4.3	3.9	2.0	—	—	.....	.....	.....	.....	.....	.....	1.3	0.6
7 a.m.	9.0	12.3	11.3	13.7	0.7	1.3	—	—	.....	.....	.....	.....	.....	.....	3.7	3.0
8 a.m.	8.4	21.0	11.8	29.4	8.1	14.6	—	—	.....	.....	.....	.....	.....	.....	6.4	8.0
9 a.m.	6.2	26.2	10.2	36.6	8.4	23.0	1.7	1.1	.....	.....	.....	.....	.....	.....	8.7	15.1
10 a.m.	3.4	27.7	7.6	43.0	7.6	31.1	3.1	3.7	.....	.....	.....	.....	.....	.....	14.0	25.0
11 a.m.	—	24.4	2.9	41.3	5.6	37.6	3.9	7.0	.....	.....	.....	.....	.....	.....	15.7	36.2
12 N.	—	—	—	33.1	2.1	42.3	4.2	11.2	3.1	1.7	—	—	—	—	16.2	48.5
1 p.m.	.....	.....	.....	.....	.....	43.4	3.0	15.1	5.8	6.2	3.9	2.2	.....	.....	15.7	60.4
2 p.m.	.....	.....	.....	.....	.....	.....	3.1	18.2	7.6	12.0	7.4	5.1	3.4	1.7	14.6	71.4
3 p.m.	.....	.....	.....	.....	.....	.....	1.7	20.7	9.4	21.0	10.4	17.4	6.2	3.6	11.5	81.2
4 p.m.	.....	.....	.....	.....	.....	.....	.....	21.6	8.1	29.4	11.8	29.6	8.4	13.7	8.4	88.6
5 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	6.7	37.6	11.3	40.1	9.0	22.4	4.8	94.0
6 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	3.9	43.3	9.1	49.5	7.6	30.6	1.7	92.5
7 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	—	44.2	—	84.1	—	34.4	—	37.1

*R_w factors are for wall conduction U-value = 0.24
R_r factors are for roof conduction U-value = 0.3

For other *wall* U-values multiply R_w factors by $\frac{U}{0.24}$

For other *roof* U-values multiply R_r factors by $\frac{U}{0.3}$

R_w factors are given for medium colored walls
For light colored walls—multiply R_w factors by 0.572
For very dark walls—multiply R_w factors by 1.285

Window Sun Effect Factor R_s

Time	N. E.		East		S. E.		South		S. W.		West		N. W.		Horizontal	
	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM
6 a.m.	116	58	125	66	52	25	.....	.....	.....	.....	.....	.....	.....	.....	10	5
7 a.m.	150	105	183	215	85	95	.....	.....	.....	.....	.....	.....	.....	.....	60	40
8 a.m.	140	240	193	400	115	330	.....	.....	.....	.....	.....	.....	.....	.....	125	135
9 a.m.	95	455	165	575	110	515	.....	.....	.....	.....	.....	.....	.....	.....	195	295
10 a.m.	45	595	110	715	90	615	10	5	.....	.....	.....	.....	.....	.....	235	505
11 a.m.	.....	630	40	780	55	680	25	20	.....	.....	.....	.....	.....	.....	260	755
12 N.	.....	.....	.....	810	10	690	30	45	10	5	.....	.....	.....	.....	270	1010
1 p.m.	.....	.....	.....	.....	.....	680	35	55	25	20	10	5	.....	.....	260	1285
2 p.m.	.....	.....	.....	.....	.....	.....	35	75	60	110	100	45	25	.....	235	1535
3 p.m.	.....	.....	.....	.....	.....	.....	30	85	110	210	165	340	95	90	195	1745
4 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	115	425	190	630	140	210	125	1905
5 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	95	530	180	805	180	355	60	1985
6 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	50	605	125	785	115	490	10	2030
7 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	.....	630	.....	830	.....	550	.....	2035

R_s factors are for unshaded windows
If awnings are used multiply R_s factors by 0.25
If Venetian blinds are used multiply R_s factors by 0.5

Window Sun Effect Factor R_s

Time	N. E.		East		S. E.		South		S. W.		West		N. W.		Horizontal	
	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM	INST	ACCM
6 a.m.	126	60	135	70	55	30	.....	.....	.....	.....	.....	.....	.....	.....	15	10
7 a.m.	145	100	185	200	100	110	.....	.....	.....	.....	.....	.....	.....	.....	65	55
8 a.m.	130	230	195	420	125	290	.....	.....	.....	.....	.....	.....	.....	.....	130	155
9 a.m.	85	440	165	585	125	445	.....	.....	.....	.....	.....	.....	.....	.....	185	315
10 a.m.	30	580	110	735	105	600	25	15	.....	.....	.....	.....	.....	.....	230	525
11 a.m.	.....	615	40	810	70	690	40	30	.....	.....	.....	.....	.....	.....	255	770
12 N.	.....	.....	.....	830	25	695	45	55	25	15	.....	.....	.....	.....	265	1030
1 p.m.	.....	.....	.....	.....	.....	685	55	80	70	65	40	20	.....	.....	255	1280
2 p.m.	.....	.....	.....	.....	.....	.....	55	170	105	150	710	95	30	15	230	1535
3 p.m.	.....	.....	.....	.....	.....	.....	35	185	125	265	165	235	85	75	185	1745
4 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	125	390	190	415	130	180	125	1905
5 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	100	505	185	625	145	320	65	2005
6 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	55	585	135	765	130	455	15	2045
7 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	.....	615	.....	835	.....	515	.....	2055

R_s factors are for unshaded windows
If awnings are used multiply R_s factors by 0.25
If Venetian blinds are used multiply R_s factors by 0.5

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Table 4-C—40 deg North Latitude Wall and Roof Sun Effect Factor R_w and R_r*

Time	N. E.		East		S. E.		South		S. W.		West		N. W.		Roof	
	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm
6 a.m.	7.9	2.9	6.7	4.5	4.8	2.3	—	—	—	—	—	—	.....	.....	3.0	0.8
7 a.m.	9.0	12.3	11.2	14.1	7.6	7.0	.....	.....	.....	.....	.....	.....	.....	.....	5.8	3.0
8 a.m.	7.9	20.7	11.5	26.2	8.7	15.7	0.6	0.2	.....	.....	.....	.....	.....	.....	6.2	8.7
9 a.m.	5.5	27.4	10.4	36.7	9.0	24.0	2.5	2.0	.....	.....	.....	.....	.....	.....	6.6	16.1
10 a.m.	2.2	31.4	7.0	45.0	8.4	33.2	4.3	5.1	.....	.....	.....	.....	.....	.....	10.2	28.4
11 a.m.	—	32.5	3.9	51.2	6.5	40.6	5.3	10.1	0.9	0.6	.....	.....	.....	.....	11.6	39.2
12 N.			—	53.2	3.9	45.6	5.6	15.7	3.0	2.2	—	—	.....	.....	11.9	47.8
1 p.m.	.....	.....	.....	.....	0.9	48.1	5.3	21.2	6.5	7.9	2.9	2.0	—	—	11.4	56.5
2 p.m.	.....	.....	.....	.....	—	48.6	4.2	26.0	8.4	16.4	7.2	7.9	2.2	1.1	10.7	63.8
3 p.m.	.....	.....	.....	.....	.....	.....	2.5	29.4	9.0	24.1	10.6	16.3	5.6	5.1	8.5	70.5
4 p.m.	.....	.....	.....	.....	.....	.....	0.6	30.2	8.7	33.0	11.9	27.7	7.9	11.8	6.2	76.9
5 p.m.	.....	.....	.....	.....	.....	.....	—	31.0	7.0	40.8	11.2	39.2	9.0	20.2	3.8	82.0
6 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	4.8	46.7	8.7	49.0	7.9	28.6	3.0	84.8
7 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	—	49.0	—	53.2	—	32.8	—	85.6

*R_w factors are for wall conduction U-value = 0.24
R_r factors are for roof conduction U-value = 0.3

For other *wall* U-values multiply R_w factors by $\frac{U}{0.24}$

For other *roof* U-values multiply R_r factors by $\frac{U}{0.3}$

R_w factors are given for medium colored walls
For light colored walls—multiply R_w factors by 0.577
For very dark walls—multiply R_w factors by 1.255

Window Sun Effect Factor R_s

Time	N. E.		East		S. E.		South		S. W.		West		N. W.		Horizontal	
	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm
6 a.m.	120	60	145	75	65	35	.....	.....	.....	.....	.....	.....	.....	.....	20	10
7 a.m.	145	200	155	240	105	120	.....	.....	.....	.....	.....	.....	.....	.....	70	60
8 a.m.	120	235	105	430	125	240	.....	.....	.....	.....	.....	.....	.....	.....	130	165
9 a.m.	70	430	165	610	140	360	25	10	.....	.....	.....	.....	.....	.....	180	225
10 a.m.	15	475	110	745	125	515	45	45	.....	.....	.....	.....	.....	.....	220	525
11 a.m.		480	40	815	90	620	65	100	.....	.....	.....	.....	.....	.....	250	780
12 N.	.....	.....	.....	825	40	660	70	170	40	25	.....	.....	.....	.....	255	1015
1 p.m.	.....	.....	.....	.....	.....	705	65	240	90	85	40	25	.....	.....	250	1270
2 p.m.	.....	.....	.....	.....	.....	.....	45	290	125	195	110	85	15	10	230	1505
3 p.m.	.....	.....	.....	.....	.....	.....	30	320	160	325	170	155	55	35	180	1705
4 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	125	445	190	415	120	105	130	1855
5 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	105	665	185	665	145	205	70	1945
6 p.m.	.....	.....	.....	.....	.....	.....	65	670	140	785	180	825	145	295	30	2030
7 p.m.	.....	.....	.....	.....	.....	.....	.....	700	.....	840	.....	890	.....	490	.....	2040

R_s factors are for unshaded windows
If awnings are used multiply R_s factors by 0.25
If Venetian blinds are used multiply R_s factors by 0.5

Table 4-D—45 deg North Latitude Roof and Wall Sun Effect Factor R_w and R_r*

Time	N. E.		East		S. E.		South		S. W.		West		N. W.		Roof	
	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm
5 a.m.	2.9	2.0	2.9	2.0	1.4	0.9	—	—	.....	.....	.....	.....	.....	.....	—	—
6 a.m.	8.1	8.1	9.3	6.7	4.6	2.9	—	—	.....	.....	.....	.....	.....	.....	1.8	1.3
7 a.m.	8.7	16.5	11.8	19.0	7.3	10.3	—	—	.....	.....	.....	.....	.....	.....	4.0	4.0
8 a.m.	7.2	24.6	11.6	30.8	9.3	18.5	1.4	0.9	.....	.....	.....	.....	.....	.....	6.1	9.1
9 a.m.	4.8	30.2	10.2	41.7	9.6	28.0	3.6	3.4	.....	.....	.....	.....	.....	.....	6.2	16.3
10 a.m.	1.4	33.9	7.6	50.4	9.0	36.7	5.3	7.5	.....	.....	.....	.....	.....	.....	9.7	25.4
11 a.m.	—	34.7	3.9	56.0	7.3	44.3	6.4	13.7	1.7	0.9	.....	.....	.....	.....	11.1	35.6
12 N.	.....	.....	.....	58.0	5.1	51.0	7.0	20.4	5.0	4.2	—	—	.....	.....	11.5	46.9
1 p.m.	.....	.....	.....	.....	1.7	54.9	6.4	27.1	7.2	10.4	—	—	—	—	11.2	57.8
2 p.m.	.....	.....	.....	.....	—	55.5	5.3	33.0	9.0	18.5	7.6	7.6	1.4	0.8	9.7	68.3
3 p.m.	.....	.....	.....	.....	.....	.....	3.6	37.1	9.8	28.0	10.2	16.5	4.5	3.9	8.3	77.4
4 p.m.	.....	.....	.....	.....	.....	.....	1.4	40.0	9.3	37.5	11.8	27.4	7.3	10.2	6.1	84.6
5 p.m.	.....	.....	.....	.....	.....	.....	—	40.8	7.3	46.8	11.5	39.2	8.7	18.2	4.0	89.6
6 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	4.6	52.1	9.3	49.5	8.1	26.6	1.8	92.9
7 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	1.4	55.1	3.9	56.2	3.9	32.5	—	93.4
8 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	—	55.7	—	58.3	—	34.4	.....	.....

*R_w factors are for wall conduction U-value = 0.34
R_r factors are for roof conduction U-value = 0.3

For other *wall* U-values multiply R_w factors by $\frac{U}{0.34}$

For other *roof* U-values multiply R_r factors by $\frac{U}{0.3}$

R_w factors are given for medium colored walls
For light colored walls—multiply R_w factors by 0.577
For very dark surfaces (roofs, etc.)—multiply above factors by 1.255

Window Sun Effect Factor R_s

Time	N. E.		East		S. E.		South		S. W.		West		N. W.		Horizontal	
	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm	Inst	Accm
5 a.m.	65	40	65	35	20	10	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
6 a.m.	135	140	130	145	70	55	.....	.....	.....	.....	.....	.....	.....	.....	20	15
7 a.m.	140	280	120	315	115	160	.....	.....	.....	.....	.....	.....	.....	.....	70	60
8 a.m.	110	405	100	505	145	280	.....	.....	.....	.....	.....	.....	.....	.....	130	165
9 a.m.	55	485	105	680	160	430	25	10	.....	.....	.....	.....	.....	.....	170	225
10 a.m.	.....	515	110	815	140	575	45	45	.....	.....	.....	.....	.....	.....	210	525
11 a.m.	.....	.....	40	890	105	700	65	100	10	5	.....	.....	.....	.....	235	780
12 N.	.....	.....	.....	910	.....	780	65	230	50	25	.....	.....	.....	.....	240	1015
1 p.m.	.....	.....	.....	.....	10	810	85	320	105	115	60	30	.....	.....	240	1260
2 p.m.	.....	.....	.....	.....	.....	815	80	385	140	225	110	95	.....	.....	215	1495
3 p.m.	.....	.....	.....	.....	.....	.....	55	445	190	380	165	225	55	25	170	1680
4 p.m.	.....	.....	.....	.....	.....	.....	.....	465	145	530	190	415	110	110	120	1800
5 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	115	660	190	630	140	225	65	1880
6 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	70	730	180	775	130	375	30	1900
7 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	20	805	65	865	65	475	.....	1900
8 p.m.	.....	.....	.....	.....	.....	.....	.....	.....	.....	810	.....	930	.....	505	.....	.....

R_s factors are for unshaded windows
If awnings are used multiply R_s factors by 0.25
If Venetian blinds are used multiply R_s factors by 0.5

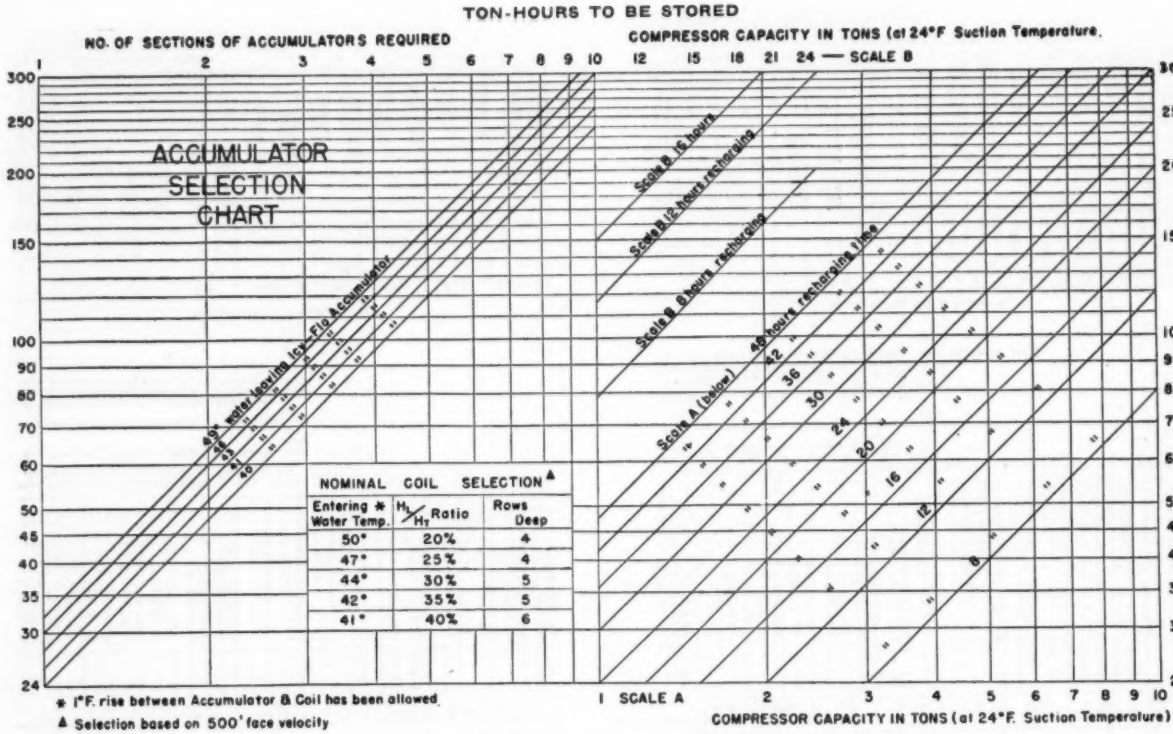
TON-HOURS TO BE STORED
NO. OF SECTIONS OF ACCUMULATORS REQUIRED
COMPRESSOR CAPACITY IN TONS (at 24°F Suction Temperature.
SCALE B
ACCUMULATOR SELECTION CHART
Scale B 1/2 hours recharging
Scale B 4 hours recharging
Scale A (below) 48 hours recharging Line
NOMINAL COIL SELECTION▲
Entering * Water Temp.
H1/H2 Ratio
Rows Deep
50° 20% 4
47° 25% 4
44° 30% 5
42° 35% 5
41° 40% 6
* 1°F. rise between Accumulator & Coil has been allowed.
▲ Selection based on 500' face velocity
SCALE A
COMPRESSOR CAPACITY IN TONS (at 24°F. Suction Temperature)

FIG. 8. ACCUMULATOR SELECTION CHART

applications will be found where this feature can be utilized to reduce electric power costs by leveling out the individual *demand* curves.

Extreme flexibility in handling a number of air conditioning zones composing one system can be secured by use of the accumulator—chilled water circulating method.

DISCUSSION

H. E. REX, Chicago, Ill. (WRITTEN): Relatively few air conditioning installations have been made utilizing stored refrigeration effect, and possibly all that have employed storage have done so through improvised equipment consisting of tanks, coils of pipe, etc. It is likely that the availability of the accumulator cell described by the authors, together with the load calculation outline, will lend impetus to the storage method for solving many air conditioning load problems.

It would be interesting to have a more detailed description of the method used by the authors for automatic control of ice thickness and for refrigerant feed. It would appear that uniformity of ice thickness during freezing should be practically self-attained due to the appreciable insulating value of ice, at least to the extent of preserving water flow passages. Fig. 2 of the authors' paper indicates that the ice is not melted uniformly during the discharge cycle.

The selection of 24 ton-hours as the capacity of one cell certainly will fit a great many applications. The writer has knowledge, however, of one installation utilizing storage involving a medium size store which would have required about 100 cells, had they been used. On this same store the load followed a weekly cycle, rather than daily. This is perhaps a factor in many applications.

The authors have selected a 6 deg rise through the circulator and suggest the same rise through the air cooling coils. No doubt this was done for convenience in illustrating the data. If the accumulator is fitted with proper baffles, the state of depletion is very nearly determined by the temperature of water returning from the air conditioner, rather than the temperature of supply water. Of course this infers storage by reason of the residual water mass, and not by ice plates. This point prompts the consideration of higher temperature rises than 6 deg at the air conditioner.

The application of appropriate safety factor is important on storage jobs to recognize the unusual day, or succession of unusual days. The exhaustion of the accumulator is rapid, after all ice is melted, due to unusual loads, leaving the user without much of anything until time for recharging is available.

The authors suggest the latent cooling period load from ventilation air be assumed constant throughout the cooling period. In the writer's opinion this leaves room for some error, particularly since this element is usually a sizable part of the total load. It would appear that some method using accumulated total heat values based on typical wet-bulb temperature variations would be in order. Similarly, internal load elements, due to occupancy, lights and appliances, sometimes deserve to be integrated as carefully as transmission and sunlight gain, when these go to make a large part of the total and when they are subject to variation from time to time during the day.

D. C. PFEIFFER, Dallas, Tex. (WRITTEN): The article is most complete and contains information which should be of general interest to all interested in air conditioning work, and of particular interest to consulting engineers, owners, and those most interested in efficient and economical operation. The application of *storage* in refrigeration work is old, but its introduction into the air conditioning field is relatively new.

Dallas, Texas, has three large office building installations of storage air conditioning, and operating records have been kept for the past three years comparing the power cost of these three buildings with other typical office buildings having conventional air conditioning systems. The results of this study confirm our early

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prediction as to the performance of this type of equipment, and certainly prove beyond a doubt that it has a decided place in the air conditioning field.

At the time these installations were being planned cold accumulators of the ice type, such as described in this article, were not available and the leading refrigeration manufacturers were invited to quote on cold accumulators consisting of large tanks containing *several miles* of pipe for the accumulation of ice. While there was nothing new in the art of freezing ice, the question of its melting capacity was uncertain and the increased cost of this type of *cold accumulators* ruled them out in favor of the so called *sweet water* tanks. The operation of these tanks consisted of a large body of city water cooled at night by a conventional water cooling system and used during normal air conditioning operation exactly as described in this article. The principal objection to this type of storage system is the large space required for the tank, and the small package units described in this article apparently answer this difficulty and also provide an efficient means of ice melting by the baffling arrangement.

The question of performance and application is so adequately covered in this article that there is little to be added. Generally speaking, *cold accumulators* should be most applicable for churches, lodge halls and restaurants for short duration of air conditioning loads and have a definite place in the air conditioning of office buildings and department stores for a normal eight to ten hours daily use.

From this article the relative investments required for this accumulator-type system as compared with a conventional direct expansion system can be obtained and possibly the only information which could be added to this discussion is the question of electric rate application and its effect upon the operating cost of the storage versus the conventional system.

Generally speaking, electric rates consist of a demand and an energy component. The particular electric service rate should be carefully analyzed and the storage equipment evaluated in terms of the actual savings in demand charge, considering the added cost for the increasing kilowatt-hours necessary in this type of system. For example, if the particular electric service rate has a demand charge of $2.50 per kilowatt per month and in the particular location approximately six months of operation constitute a normal season, then there will be a yearly saving in demand charge of six times $2.50 or $15.00 for each kilowatt less than the conventional system. If this particular application has an energy cost of 1 cent per kilowatt-hour (in the particular step of the rate) the storage system should be charged with an estimate of the additional kilowatt-hours required over the conventional system at 1 cent per kilowatt-hour.

It is apparent from this discussion that the higher the demand charge and the lower the energy charge the greater will be the justification for storage installations. A general rule cannot be given except to compute a daily heat gain curve, thus determining the hours of on-peak and off-peak operation. From this information, together with a study of the electric rate, the true economics of storage installations can be determined.

S. R. LEWIS, Chicago, Ill. (WRITTEN): This paper gives intelligible and helpful suggestions to any one needing data on the undoubtedly beneficial process of accumulating a reserve of refrigerating effect during off-peak hours. In many air conditioning plants, particularly for theaters and churches, a reduction in the investment and operating costs is practicable by operating the condensing unit longer hours and thereby reducing the size and the power demand charge. Where a large reservoir of chilled water is employed the problems of space occupied, of insulation and of baffling the water within the storage tank are often serious.

This paper provides a background for the whole computation and the accumulators appear to be very compact and are easily insulated.

I think the new terms *accumulated degree hours* and *accumulated sun intensity values* are a contribution to simplification of the rather abstruse thinking which must accompany a design for stored refrigerating effect.

K. C. PORTER, Chicago, Ill. (WRITTEN): The authors of this paper had the rare opportunity of presenting a device which is interesting and useful during ordinary peace time conditions but is doubly useful in the refrigeration and air conditioning field during war time conditions. Since the preparation of this paper much of our production machinery has been put on a 24-hour basis. In the cold accumulator we have a device which can extend the usefulness of our present refrigeration and air conditioning equipment to 24 hours a day.

The equipment herein described calls for copper plate coils. I have been advised that performance characteristics with steel plate coils are entirely satisfactory. Therefore, equipment production should not be restricted by defense (critical material) needs. Since power restrictions may be expected to be concerned largely with system peak demands, the longer hour use of existing refrigeration and air conditioning equipment should be tolerated because of its off-peak nature.

Section II of this paper which deals with performance characteristics proved to be technically disturbing. For instance, we read that the total cooling effect of this 24-ton accumulator varied from 24 to 33 tons depending upon the final water temperature. Since we ordinarily consider a storage battery and an ice accumulator to have a specific amount of available energy, some explanation of the paradox should be given by the authors.

In first paragraph on p. 127, the statement is made that "the total ton-hour output of one such accumulator is dependent only upon the temperature of the water to be delivered by the accumulator, regardless of the total water circulation required." On p. 126, in paragraph immediately above Fig. 2 there is an analysis where one accumulator delivering 40 deg water has a capacity of 25¼ ton-hours and two accumulators delivering 40 deg water have a capacity of 26½ ton-hours each.

It is my belief that much of the reader discomfort with this paper can be traced to the manner in which the test data were obtained. Although it is recognized by the authors that the air conditioning job must be designed for a fixed water temperature (between 40 deg and 50 deg) the test data are taken without attempt to maintain constant discharge temperature. It would seem to me that test results obtained with the use of a three-way temperature regulating valve would be much more convincing.

Section III, Calculation of Total Cooling Load—offers some interesting material on the accumulation method. It is unfortunate, however, that the authors' material does not check through with better accuracy. For instance, on p. 135 the authors give an example of the use of tables for the determination of accumulated sun load. Their example shows for wall and roof factors, accumulated R_w at noon to be 52.2 for east exposure and 6.4 for south exposure. The table for 30 deg north latitude (Table 4A) shows these noon values to be 52.1 for east exposure and 6.5 for south exposure. Although these errors are inconsequential, they are disturbing to the reader. Since they disrupt the continuity of the reader's thought and harm his confidence in the material, such errors should be avoided.

In conclusion, it is my feeling that the cold accumulator has two markets in the war air conditioning picture.

1. *Defense Industry:* Here some economy in size of refrigeration machines and peak load power demands may be achieved by designing for average rather than maximum cooling loads. However, as our defense industries go on a 24-hour basis, the economy of the accumulator becomes questionable.
2. *Civilian Comfort:* As the defense program has been taking hold, our ordinary commercial comfort market is taking a rain check for the duration. For the civilian prospect who has been denied a 20-ton full capacity installation, the 10-ton installation (with accumulator) will possibly also be denied, even though his maximum load upon his utility might be halved in this manner.

The greatest usefulness for this type of equipment *for the duration* appears to be in extending the total cooling capacity of present comfort and industrial installations by a 24-hour ice machine operation.

JOHN HOWATT, Chicago, Ill. (WRITTEN): This paper is a timely presentation of factual data on an addition to the equipment available for use in the summer air

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conditioning field. Cold accumulators have been used in refrigeration work almost from its beginning, in fact the business of the ice man was based entirely upon his ability to distribute accumulated cooling effects. Cold accumulators are now used in connection with air conditioning work but not as extensively or with the same effectiveness as will be possible with the development of a commercial type of accumulator such as described in this paper.

The purpose of any cold accumulator is to provide cooling effects in off-peak times when the compressor is shut down. The objective should be to, as far as possible, level out the hills and valleys in the compressor load making possible the use of such smaller compressor equipment, equipment that can be used at nearly full load capacity 24 hours per day. On most of the direct expansion jobs in air conditioning at present the compressor unit or units are designed to take care of the peak cooling load conditions which may occur for only a few hours and which is always very much greater than the average 24-hour load. The same condition prevails in the selection of boilers to take care of the heating load in winter. The heating equipment, for example, must be designed to take care of outdoor 10 deg below zero temperature, whereas the average for the coldest 24 hours would always be higher. In heating load calculations but little cognizance is taken of the fly wheel action of the building structure in that it serves as a type of heat accumulator.

The design of the cold accumulator in unit form lends itself readily to arrangements to meet various capacity requirements so is flexible enough to give the designer sufficient latitude. It also makes it possible for the designer to adapt the units to available space and should appeal to engineers, contractors and owners.

It would seem from a study of the paper that the accumulators as designed are possible of reasonably accurate rating under prescribed conditions which feature will make it possible for the designer to make his selection from the manufacturers rating tables with the assurance that the performance of equipment in the field will be equal to their published ratings. This is important in the selection of all engineering equipment today.

On the whole it appears the development is timely in our whole national conservation program at this time because the use of cold accumulators will save time, material and money in the manufacture and installation of air conditioning systems.

No. 1198

THE EFFECT OF ATTIC FAN OPERATION ON THE COOLING OF A STRUCTURE

By W. A. HINTON * AND A. F. POOR,** ATLANTA, GA.

This paper is the result of research sponsored jointly by the Atlanta Chapter and the Committee on Research of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the State Engineering Experiment Station at the Georgia School of Technology.

INTRODUCTION

DURING the summer months the minimum outside air temperature at night is considerably lower than the maximum temperature of the preceding day. Data from the U. S. Weather Bureau indicates that in Atlanta, Ga., this temperature difference averages approximately 18 F. The outside temperature usually starts to drop in the afternoon and the decrease is quite rapid during the early evening hours. By 10 P.M. approximately two-thirds of the temperature drop has occurred.

A house and its contents can be cooled during the late afternoon and early evening by drawing through it the cooler outside air. This permits the walls, ceilings and floors to transfer heat from both inside and outside surfaces, reducing the amount of heat that must be conducted through the building materials to the outside surface. The flow of the cool air increases comfort conditions in the house during the cooling period.

In recent years attic fans have been installed in a number of houses in this section of the country. These fans are mostly of the axial or propeller type and they are usually installed so that outside air is drawn in through the house and discharged from the attic. Various arrangements have been used in making the installations.

With the increasing use of these fans a need was felt for further study of their performance. Through the cooperation of the Atlanta Chapter and the Committee on Research of the ASHVE, this investigation was undertaken

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TABLE 1—STRUCTURAL AND THERMAL PROPERTIES OF THE HOUSES

	TYPE OF CONSTRUCTION	COEFFICIENT OF HEAT TRANSMISSION U
Floor	Pine on pine with building paper between	0.34
Outside Wall	2 in. × 4 in. Studs on 16 in. centers inside ½ in. sheet rock, outside ⅞ in. lap siding on ⅞ in. pine sheathing	0.25
Roof	Composition shingles on ¾ in. pine	...
Ceiling	½ in. sheet rock. 2 in. × 6 in. joists on 16 in. centers	0.437[a]

[a] Combined roof and ceiling coefficient.

to determine methods of fan selection, operation and installation which would bring about a maximum cooling of the structure and thereby afford the greatest comfort to the occupants consistent with economy of installation and operation. It was also desired to obtain data by which cooling effects of the fan could be compared with the natural cooling of a house. This paper describes the method used to obtain this information and contains preliminary data obtained from test runs made during the summer of 1941.

A considerable amount of information has been compiled by Badgett on the installation and use of attic fans.[1] Much of this information is of a general nature regarding structural details and methods of operation. Some cost data were also included.

The use of night air for cooling was investigated by Kratz and Konzo.[2] The night air was used to reduce the daytime cooling load on a cooling system in the Research Residence at the University of Illinois. Outside air was drawn through the house at night while during the daytime the house was kept closed. With this arrangement requirements for artificial cooling were decreased.

Further information on summer cooling studies carried on at the Research Residence has been reported also by Kratz and his co-workers.[3]

TEST EQUIPMENT AND ITS ARRANGEMENT

In order to determine more accurately the cooling effect of an attic fan it was decided that temperature measurements should be made in two identical houses located next door to each other. Two such houses were found and a fan was installed in one of them. Thermocouples were installed in each of the rooms of both houses. The house without a fan, which will be referred to as the control house, was used to determine the cooling effects caused by

[1] The Installation and Use of Attic Fans, by W. H. Badgett. (*Bulletin*, Agricultural and Mechanical College of Texas.)

[2] Study of Summer Cooling in the Research Residence for the Summer of 1933, by A. P. Kratz and S. Konzo. (ASHVE TRANSACTIONS, Vol. 40, 1934.)

[3] Study of Summer Cooling in the Research Residence for the Summer of 1934, by A. P. Kratz, S. Konzo and M. K. Fahnestock. (ASHVE TRANSACTIONS, Vol. 41, 1935.)

natural circulation. The added cooling effects of the fan in the test house could, therefore, be calculated.

THE HOUSES

The two houses selected are single story frame structures possessing identical floor plans. They are the same size and are constructed of the same kind of materials. The houses rest on piers and the floors are approximately 18 in. above the ground. The distance between houses is approximately 12 ft. Neither house has a basement. Details of construction are tabulated in Table 1.

Each house has a volume of 8,900 cu ft. Volumes of individual rooms and the areas of floors, windows and door openings are tabulated in Table 2. The windows and doors are not weather stripped and neither house has awnings over the windows.

The floor plans of the two houses are shown in Fig. 1. Rooms in corresponding positions in the two houses were used for similar purposes. Fig. 1 also shows the location of the thermocouples and the location of the intake grille of the fan in the test house.

Both houses were occupied during the tests. The test house was occupied by two adults. The control house was occupied by three adults.

THE FAN

The fan installed in the test house is an axial type, 36 in. in diameter and fitted with four blades. It is rated to deliver 10,000 cfm at a speed of 350 rpm and 0.01 in. water gage static pressure. The arrangement of the fan installation is shown in Fig. 2. The opening in the ceiling was covered with an expanded metal grille having 70 per cent free area. The grille opening is 5 ft long and 3 ft 10 in. wide. Two ceiling joists run lengthwise of the opening, reducing the gross area to 17.5 sq ft.

With the fan installed in this manner the outside air could be drawn in through the windows and then through the rooms and the hall and finally through the grille opening. The fan discharged the air into the open attic space. A louvred opening in the gable supplemented by openings placed in the eaves enabled the air to escape from the attic to the outside.

TABLE 2—DIMENSIONS OF ROOMS AND OPENINGS IN THE HOUSES

ROOM	ROOM NO.	VOLUME CU FT	FLOOR AREA SQ FT	WINDOWS[a]	DOORS[b]
Living room	I	1935	193.5	2	2
Front bed room	IV	1950	195.0	2	2
Rear bed room	V	1420	142.0	2	2
Kitchen	II	2200	220.0	2	2
Hall	III	1390	139.0	0	2
Total	...	8895	889.5	8	10

[a] Wood sash double hung 33 × 72 in.
[b] Wood panel 3 × 7 ft.

An anemometer, range 200 to 2000 fpm, was used to measure the rate of air flow. This instrument was calibrated prior to its use in the tests. The calibration was made for this particular use. The fan was set up in the laboratory and equipped with the same intake box and grille that was to be used in the test house. The fan intake box and duct are shown in Fig. 3. A duct with pitot tube stations was attached to the discharge side of the fan. By placing different size orifices on the discharge end of the duct, varying rates of air flow were obtained. With the fan in operation, pitot tube readings were taken according to the Standard Test Code Procedure of the *National Association of Fan Manufacturers.* At the same time anemometer readings

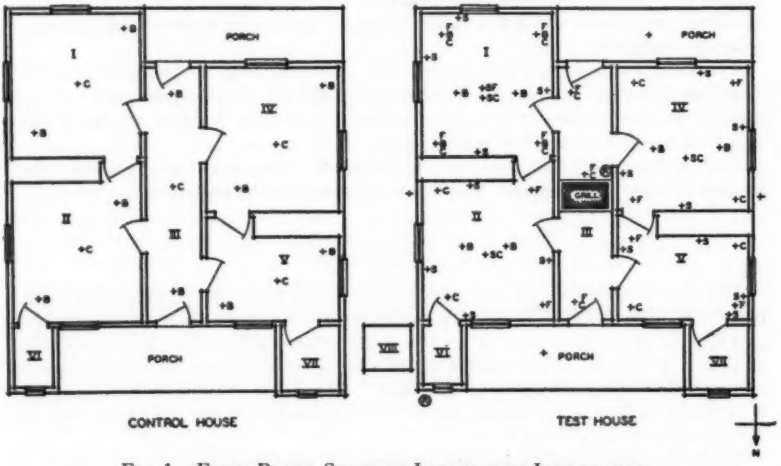

FIG. 1. FLOOR PLANS SHOWING LOCATION OF INSTRUMENTS

Key: I Living Room; II Kitchen; III Hall; IV Front Bed Room; V Rear Bed Room; VI Pantry; VII Bath Room; VIII Instrument Booth; R Location of Temperature Recorder Bulbs. Thermocouple locations are indicated by marks followed by letters indicating levels of positions, as follows: F Floor Level; B Breathing Level; C Ceiling; S Surface Temperature; SC Surface of Ceiling; SF Surface of Floor.

were taken under the grille at 15 stations. The average values of the air velocities at the grille were plotted against corresponding volumes in cubic feet per minute computed from the pitot tube readings. The curve obtained (Fig. 4) was used in computing the rates of air flow from anemometer readings taken during the test runs.

THE THERMOCOUPLES

In the two houses 90 thermocouples (iron constantan) were installed. They were checked for accuracy by comparison with a mercury in glass thermometer

which had been calibrated by the U. S. Bureau of Standards. Thermocouples were placed in each room and in the hall of the test house in the following positions: Breathing level; floor level; and ceiling. Horizontal distribution was used at each level. In addition, thermocouples were placed in contact with the walls and ceilings of each room so that surface temperatures could be measured. In the control house thermocouples were located at various levels in all rooms and in the hall. A few thermocouples were also placed in the attic, on the roof and just outside of both houses. The thermocouple locations are shown in Fig. 1.

The leads from all thermocouples were run to an instrument booth built between the two houses. They were connected through switches to a potentiometer. Fig. 5 shows the arrangement of switches and potentiometer in the instrument booth.

In order to supplement the thermocouple readings, two temperature recorders were installed. These recorders were kept in adjustment to give readings

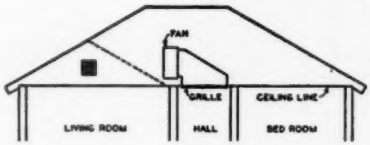

FIG. 2. FAN INSTALLATION IN TEST HOUSE ATTIC

the same as the calibrated thermometer. One recorder was set up in the hall of the test house. A record of the average inside air temperature could thereby be obtained. The other temperature recorder was installed in the instrument booth with the bulb located to give outdoor air temperatures.

OPERATING PROCEDURE

Each test run involving the operation of the attic fan was carried out according to the following general outline. Just before the fan was turned on a set of thermocouple readings was taken. A set of readings consisted of one reading for each of the 90 thermocouples installed in both houses. The temperatures were recorded to the nearest ½ F. The windows and doors were opened or closed as desired and then the fan was turned on. Reading of the thermocouples was continued and complete sets of observations were made at approximately 30-min intervals during the run. The rate of air circulation for each run was determined by taking 15 anemometer readings under the grille and then using the anemometer calibration curve.

A number of variations were made on the above procedure. The time of starting the fan was varied from 12 noon to 5 P.M. During some runs the fan was turned off at approximately 9 P.M., while on others it was permitted

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to operate until early the next morning. The front and rear entrance doors were always closed during a run. On some of the runs one or more of the rooms was completely closed off. In such cases the volume of the open rooms through which the air was circulated was used to compute the number of air changes per hour. The amount of infiltration of air drawn through the closed rooms can be considered as negligible compared with the total volume of air handled by the fan.

One 24-hour run was made to compare the test house and the control house from the standpoint of their heat capacity, measured under normal air circulation conditions. In this run the grille opening to the fan and all the openings from the attic of the test house were closed off. The fan was not

FIG. 3. THE FAN AND DUCT AS SET UP IN THE LABORATORY

turned on. The door and window openings were identical in both houses. Thermocouple readings were taken every hour during the 24-hour period.

The personnel available did not permit the reading of the thermocouples to be continuous over 24-hour periods except during the one run mentioned above. The recording thermometers provided 24-hour records of the test house and outdoor temperatures. Since all the air drawn through the rooms of the test house entered the hall and was then drawn up through the fan, the temperature indicated and recorded by the thermometer placed in the hall was an average value of the temperatures of the rooms. A comparison of the various open room thermocouple readings with the readings of the temperature recorder in the hall shows that they are in close agreement (Fig. 11). Weather observations were recorded qualitatively.

RESULTS AND DISCUSSION

During July, August and September, 30 test runs were made. The results from typical runs are presented in the tables and figures. The period of fan

operation; the number of air changes per hour; and the temperature data are indicated on each curve. The temperature indicated for a particular room is the average of the readings of all breathing level thermocouples in that room. The average inside air temperatures of the test house, which are plotted on a number of the curves, were taken from the hall recorder charts.

The nature of the data recorded during each of the test runs is indicated in Table 3, which presents in tabulated form the various temperatures observed in both houses during a typical run. Attention is called to the fact that during this run the living room and kitchen windows were closed at 7 P.M.

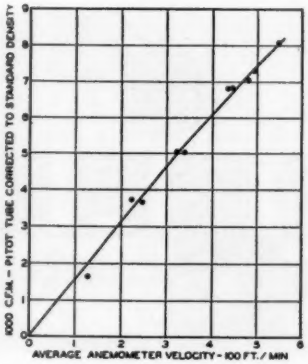

FIG. 4. CALIBRATION CURVE OF ANEMOMETER

All of the data obtained from the test runs have not been tabulated in this paper. Instead, most of the data presented are in the form of curves. Presentation of the results in this form simplified the anaylsis and the interpretation of the data.

During the runs indicated by Figs. 6 and 7 the living room of the test house was closed. The windows and doors of all the other rooms were open. In the run shown by Fig. 8 the living room and the kitchen were closed off.

Comparison of Figs. 6, 7, and 8 shows the effects of starting the fan at different times of the day. Fig. 8 shows that the outside temperature had reached its maximum and had started to decrease before the fan was turned on. It will be noted that within a few minutes after the fan was turned on the inside air temperature started to drop rapidly.

From Figs. 6, 7, and 8 it can be seen that the slope of the inside temperature curve approaches the slope of the outside temperature curve during the period of fan operation. With the fan in operation, the slopes of the two curves become practically equal within one to two hours after the outside

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temperature drops down to the inside temperature. When the slopes of the two curves are equal the rate of decrease in inside temperature is the same as that of the outside temperature.

From these curves it can be concluded that the fan should be turned on just as soon as the outside temperature drops down to the temperature on the inside of the house. During the summer months in Atlanta this time ranges from 4 to 5 P.M. It will be noted from all the curves presented that within two hours after starting the fan (provided the outside air temperature is dropping) the inside temperatures drop to within two or three degrees of the outside temperatures. After three to four hours of fan operation this temperature differential is less than two degrees. Fig. 9 shows among other things

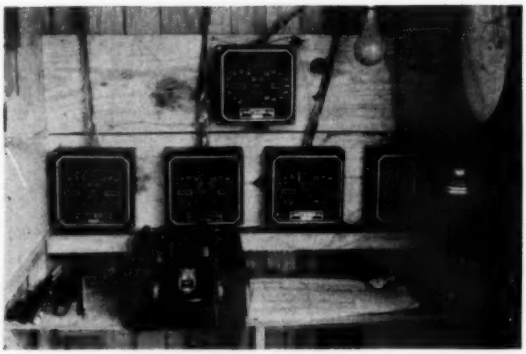

FIG. 5. THE INSTRUMENT BOOTH SHOWING SWITCHES AND POTENTIOMETER

that within one hour after turning on the fan during late afternoon the test house temperatures start dropping much more rapidly than the control house temperatures. It can further be seen that even five hours later the control house temperatures have dropped on an average of only three degrees. During this same time interval the outside air temperature has dropped approximately 10 deg and the average test house temperature has been lowered approximately eight degrees.

Figs. 8, 9, and 11 represent runs in which the fan was started at 5 P.M. and stopped at 7 A.M. the next day, 10 P.M. and 12 midnight, respectively. The inside and outside temperature curves for the test house indicate that within an hour after the fan is turned off the inside temperatures do not continue to drop at the same rate as the outside temperatures. Instead, the inside temperatures actually rise slightly (up to 1 deg) during this time interval and then start dropping again but at a much lower rate. This slight temperature rise is caused by the transfer of heat from the warmer walls and ceiling of the structure to the substantially still air in the house.

From the data presented it can be concluded that a one story frame structure, of low heat capacity, can be brought down to within a few degrees of the

TABLE 3—TABULATION OF DATA TAKEN ON TYPICAL RUN

TEST HOUSE

TIME OF DAY.....		3:25	4:05	5:00	5:30	6:30	7:30	7:55	8:20	8:55	9:25	9:55
Living Room Temp.	Floor	86.5	85.9	86.8	86.8	86.6	85.0	84.8	84.6	84.5	83.5	83.5
	B. L.	89.3	90.2	91.0	88.9	87.5	85.5	85.7	85.5	84.5	84.5	84.0
	Ceil.	92.0	91.3	93.0	87.0	90.0	87.5	86.5	86.5	85.0	85.0	84.0
Front Bed Room	Floor	87.5	91.2	89.5	87.8	85.5	82.8	82.5	81.8	81.2	80.0	79.0
	B. L.	89.0	91.8	90.5	89.0	85.5	83.5	82.5	81.2	81.0	79.5	78.5
	Ceil.	90.0	92.0	91.0	90.5	86.8	83.8	82.5	81.8	81.5	80.2	79.5
Rear Bed Room	Floor	85.8	89.2	88.5	87.8	85.5	82.8	82.0	81.8	81.0	80.0	79.2
	B. L.	87.0	91.5	89.5	87.8	86.5	84.0	83.2	82.0	81.5	80.5	79.5
	Ceil.	87.8	92.0	90.5	89.8	87.2	83.2	83.0	82.0	81.2	80.5	79.5
Kitchen Temp.	Floor	86.0	89.0	89.0	88.0	85.0	84.2	84.5	84.5	84.5	83.5	82.5
	B. L.	88.5	90.5	90.0	88.2	85.5	85.5	85.5	84.5	84.5	84.0	82.5
	Ceil.	90.8	91.8	91.5	90.2	87.0	85.2	85.5	84.5	84.5	83.5	82.5
Hall Temp.	Floor	85.8	89.0	87.8	87.5	84.7	83.2	82.2	81.8	81.0	79.8	79.7
	B. L.					None						
	Ceil.	87.3	91.2	89.5	89.0	86.2	84.7	83.0	82.5	81.3	80.7	80.5
Avg. Temp. Under Floor.....		83.0	84.5	84.0	84.2	80.8	80.5	80.0	79.8	80.0	78.8	78.5
Avg. Outside Temp.		91.0	91.2	88.0	96.2	82.2	81.2	81.5	80.2	79.8	78.2	76.0
Avg. Attic Temp....		106.2	108.5	94.2	92.4	88.3	85.7	84.4	83.5	82.4	81.3	80.6

SURFACE TEMPERATURE

Living Room.....		89.1	90.3	89.7	88.8	87.6	85.8	85.2	85.1	84.4	83.4	82.7
Front Bed Room....		90.0	91.3	90.6	89.3	87.3	85.3	84.2	83.2	82.7	81.6	81.1
Back Bed Room....		88.6	91.1	90.4	90.2	87.7	85.2	84.0	83.2	82.8	81.6	81.0
Kitchen.....		89.1	90.3	89.5	88.4	86.5	85.5	85.2	84.2	84.0	82.7	82.3

CONTROL HOUSE

Living Room	B. L.	92.0	89.2	92.5	93.0	...	90.5	89.0	87.2	87.5	86.8	87.5
	Ceil.	95.0	93.5	94.5	94.0	...	92.0	89.5	87.5	82.5	87.0	89.0
Fr. Bed Room	B. L.	91.0	91.8	92.2	93.0	...	91.0	90.2	88.2	88.2	87.8	87.8
	Ceil.	95.0	95.0	96.0	95.0	...	92.5	92.0	91.0	90.0	89.5	89.0
Back Bed Room	B. L.	88.5	91.5	91.0	91.8	...	90.5	89.5	88.5	88.2	87.2	87.5
	Ceil.	93.5	95.0	95.0	95.0	...	92.5	92.0	91.0	90.0	89.0	89.0
Kitchen Temp.	B. L.	93.8	89.2	90.0	92.5	...	90.5	89.5	88.2	88.8	87.8	87.5
	Ceil.	97.0	94.0	94.0	95.0	...	92.5	91.5	91.5	90.0	89.5	88.5
Hall Temp.	B. L.	88.5	90.5	88.8	92.8	...	90.8	89.5	86.5	87.8	87.5	87.8
	Ceil.	88.0	92.0	92.0	94.0	...	92.5	92.0	88.0	90.0	89.0	89.0
Avg. Attic Temp....		108.5	108.7	109.0	107.2	...	98.8	96.8	95.5	94.0	91.5	90.2

B. L. = Breathing Level.
Fan in operation from 4:50 to end of run.
Dark at 7:20.
Living room and kitchen in test house closed off.
Air changes per hour, 80.
All temperatures are averages of several thermocouple readings, degrees Fahrenheit.
Date of run, September 2, 1941.

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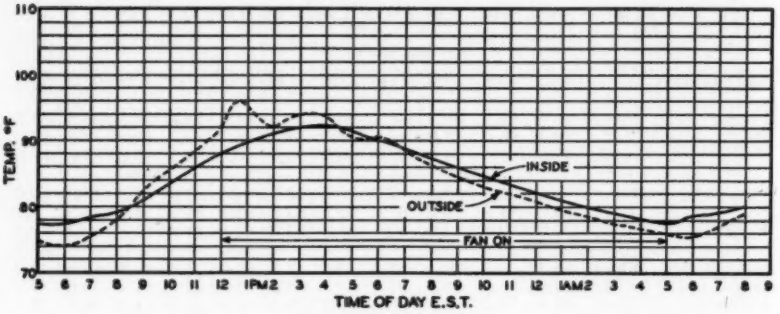

FIG. 6. RELATION OF TIME AND AVERAGE INSIDE AND OUTSIDE TEMPERATURES (55 AIR CHANGES PER HOUR; JULY 30-31, 1941; WEATHER, PARTLY CLOUDY)

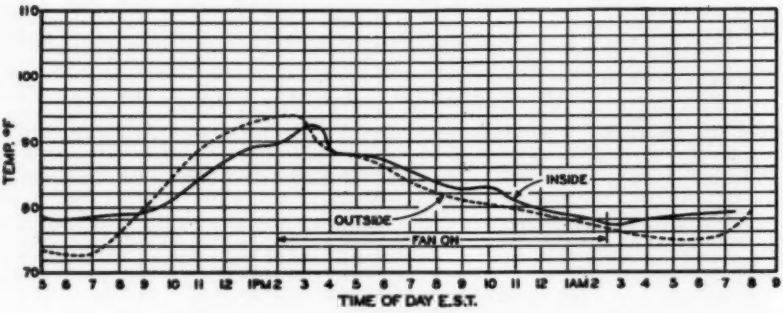

FIG. 7. RELATION OF TIME AND AVERAGE INSIDE AND OUTSIDE TEMPERATURES (57 AIR CHANGES PER HOUR; AUGUST 9-10, 1941; WEATHER, CLEAR)

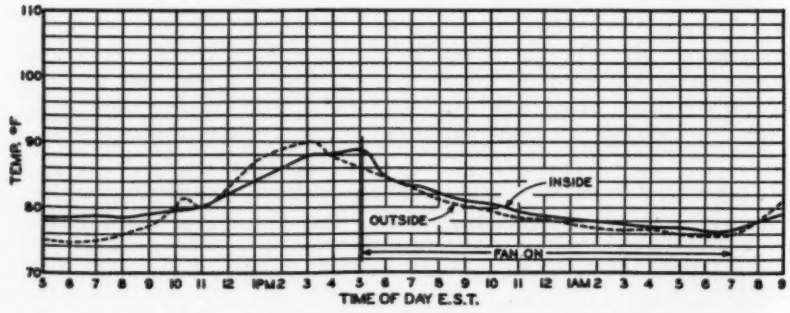

FIG. 8. RELATION OF TIME AND AVERAGE INSIDE AND OUTSIDE TEMPERATURES (80 AIR CHANGES PER HOUR; SEPTEMBER 3-4, 1941; WEATHER, CLEAR)

existing outside air temperature if the fan is operated for only three or four hours and that further cooling will occur, although at a lower rate, after the fan is turned off. The optimum time for turning off the fan would depend on the actual outside temperatures and on the individual personal preferences. In a better insulated structure the further cooling characteristics might be of a different nature. If the maximum lowering of the temperature of the inside air and of the structure itself is desired, the fan should be operated until approximately 6 A.M. the following morning. By following

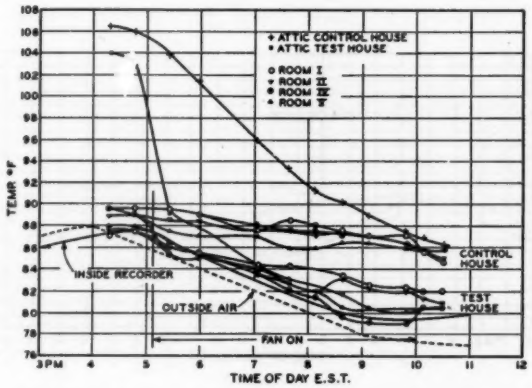

FIG. 9. RELATION OF TIME AND INDIVIDUAL ROOM TEMPERATURES IN TEST AND CONTROL HOUSES (60 AIR CHANGES PER HOUR; SEPTEMBER 10-11, 1941; WEATHER, PARTLY CLOUDY)

this procedure of all night fan operation the house would remain cooler a longer period of time during the following day, especially if the doors and windows were closed.

The effects of varying the number of air changes per hour on the temperatures inside the test house can be observed from Figs. 6, 7, 9, 11 and 12. A minimum of 45 air changes per hour (calculated) were used during this entire investigation. A larger number of air changes per hour were accomplished by closing off some of the rooms in the test house. The curves indicate that the differential reached between inside and outside temperatures, after a given number of hours of fan operation, was not measurably affected by increasing the rate of air changes above 45 per hour. The inside temperatures referred to here are for the rooms which were open.

In the run shown by Fig. 9, room I was closed at 7:00 P.M. while room II was closed at 8:15 P.M. From these curves it is seen that these two rooms do not cool as rapidly as the ones left open. The closing of rooms I and II increased the number of air changes in the rooms left open. Fig. 9 shows

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that the effect of increasing the rate of air flow did not produce any significant change in the rate of cooling of the rooms that were left open.

These results indicate that the use of air changes in excess of 45 per hour would be of little benefit so far as further cooling of a typical one story frame structure is concerned and would, therefore, be uneconomical. Further investigation may show that even less than 45 air changes per hour will be sufficient.

A comparison of the temperatures in corresponding rooms of the test house and the control house points out several advantages of the fan operation.

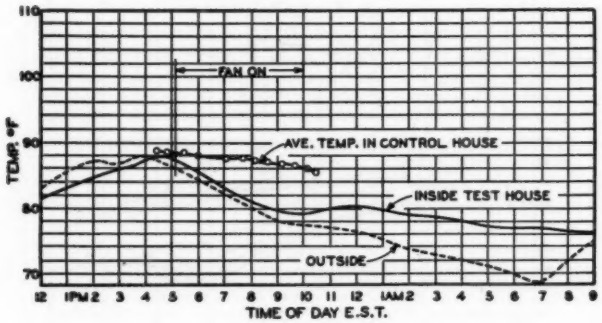

FIG. 10. RELATION OF TIME AND AVERAGE INSIDE TEMPERATURES OF TEST AND CONTROL HOUSES (60 AIR CHANGES PER HOUR; SEPTEMBER 10-11, 1941; WEATHER, PARTLY CLOUDY)

Referring to Figs. 9 and 10, it can be seen that, with the fan in operation, the test house temperatures start decreasing just after the outside temperature drops below existing inside air temperatures, while there is a lag of one to two hours before the control house temperatures begin to drop. From Fig. 9 it can be seen that after several hours of fan operation, the individual rooms in the test house are kept five to eight degrees cooler than the corresponding rooms in the control house. It can also be noted that the temperature in the test house attic drops down very close to the temperature of the rooms soon after the fan is turned on, while the attic of the control house cools over a much longer period of time. The removal of this body of hot air from the attic space will obviously make the space below easier to cool.

Fig. 13 shows a comparison of average breathing level and surface temperatures for two rooms of the test house maintained under different conditions. With the fan in operation, the living room was kept closed while the other rooms were open. The surface temperatures in the living room were found to be slightly lower than breathing level temperatures. This indicates that some heat was being transferred from the air to the walls and ceiling. In the bed room, which was open and through which air was drawn, the sur-

face temperatures were found to be higher than the inside air temperature, indicating that heat was being transferred from the surfaces to the air. The rate of decrease of surface temperatures was found to be very close to the rate of decrease in breathing level temperatures, as can be seen from Fig. 13.

A 24-hour test run was made to compare the temperatures of the test house and the control house when the fan was idle. The results of this run are shown in Fig. 14 in which the average breathing level temperatures are plotted for each house. These temperatures were observed hourly by taking thermocouple readings.

From the curves it can be seen that the temperatures in the control house were approximately two degrees higher than the test house temperatures,

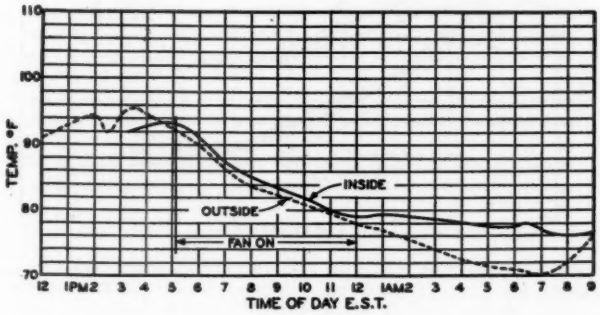

FIG. 11. RELATION OF TIME AND AVERAGE INSIDE AND OUTSIDE TEMPERATURES (80 AIR CHANGES PER HOUR; SEPTEMBER 9-10, 1941; WEATHER, CLEAR)

except during the cooking periods, when the temperature difference was greater, and during the period from midnight until 5 A.M., when the temperature difference was approximately one degree.

The higher control house temperatures can probably be attributed to the fact that a kerosene stove was used for cooking purposes while an electric stove was used in the test house.

ATTIC OPENINGS FOR EGRESS OF AIR

At the beginning of this investigation, the louvred opening in the gable of the test house provided only two square feet of flow area for the exit of air from the attic. The resistance of air to flow through this small opening reduced the fan delivery. Other openings were placed in the eaves of the house to make a total of eighteen square feet. This increase in area of opening resulted in an increase in fan delivery from 6000 cfm to 7000 cfm.

After this an additional opening was placed in the attic and no appreciable increase in fan delivery was obtained. Since the gross area of the inlet grille

was 17.5 sq ft, it can be concluded that the total area of the openings from the attic should be approximately equal to the gross area of the inlet grille.

SUMMARY

An investigation has been made on the operation of an attic fan in a single story frame structure and the temperatures produced have been compared with those obtained at the same time from an identical structure not equipped with a fan.

The following conclusions can be drawn from the results obtained during thirty test runs, conducted during the summer of 1941, in which operating conditions and time intervals were varied:

1. When outside air is drawn through a typical single story frame structure at a rate of 45 air changes per hour, the inside air temperatures are lowered to within two degrees of the existing outside air temperature.
2. With 45 air changes per hour the differential between inside and outside temperatures is maintained effectively constant. The slopes of the curves for inside and outside temperatures are practically equal.
3. An increase in the number of air changes above 45 per hour does not bring about any appreciable reduction of this temperature differential, nor does it increase the slope of the indoor temperature curve.
4. The optimum time in Atlanta, Ga., for starting the fan was found to range from 4 to 5 p.m.
5. After the fan has been in operation for three or four hours in the evening it can be turned off provided the inside temperature has been lowered sufficiently unless continued operation is desired by the occupants to produce pre-cooling for the following day.
6. The test house was found to be five to eight degrees cooler than the control house.
7. The total area of the openings in the attic for egress of air should not be less than the gross area of the intake grille.

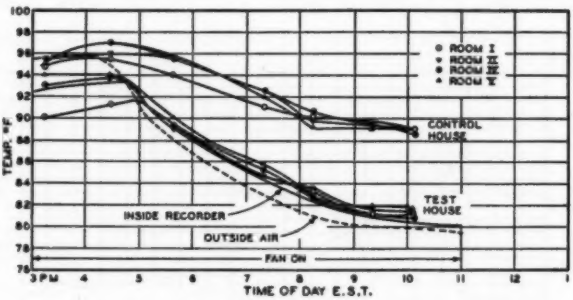

FIG. 12. RELATION OF TIME AND INDIVIDUAL ROOM TEMPERATURES IN TEST AND CONTROL HOUSES (46 AIR CHANGES PER HOUR; AUGUST 11-12, 1941; WEATHER, CLEAR)

OUTLINE OF FUTURE WORK

1. Investigation of effect of lowering number of air changes per hour.
2. A further study to establish effective temperatures in both the test house and

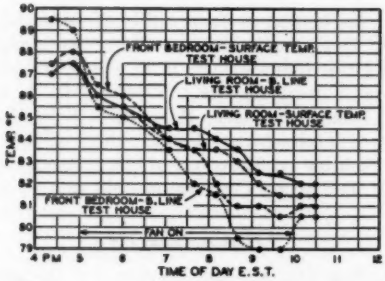
FRONT BEDROOM - SURFACE TEMP. TEST HOUSE
LIVING ROOM - B. LINE TEST HOUSE
LIVING ROOM - SURFACE TEMP. TEST HOUSE
FRONT BEDROOM - B. LINE TEST HOUSE
FAN ON
TEMP. °F
TIME OF DAY E.S.T.

FIG. 13. RELATION OF SURFACE TEMPERATURES TO BREATHING LEVEL TEMPERATURE (60 AIR CHANGES PER HOUR; SEPTEMBER 10, 1941; WEATHER, PARTLY CLOUDY)

the control house. (Some preliminary data, not included in this paper, were obtained during the present investigation.)

3. A study of attic fan installations discharging air directly to the outside, with attic as an inlet plenum.

4. A study of attic fan installations which will exhaust the house only at night and the attic only during the day.

5. A study of the resistance to air flow in typical attic fan installations including a comparison of installed delivery with free delivery.

6. Investigation of the combined effect of insulation and attic fan operation.

7. Investigation of attic fan operation in multi-story houses and in residences of different types of construction.

ACKNOWLEDGMENTS

The authors wish to express their appreciation for the cooperation of J. A. Miller, president and general manager of the Exposition Cotton Mills, Atlanta, Ga., who made possible the use of the two houses. Further acknowledgment

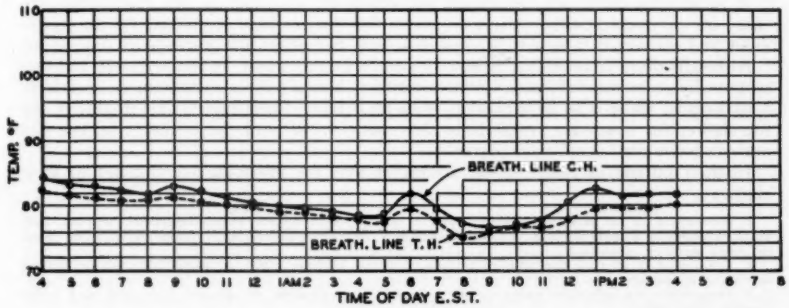

FIG. 14. RELATION OF TEMPERATURES IN TEST AND CONTROL HOUSES WITH FAN IDLE (SEPTEMBER 24-25, 1941; WEATHER, CLOUDY, PRECIPITATION 0.36 IN.)

should be made for the courteous cooperation of Mr. and Mrs. J. A. Towns and Mr. and Mrs. A. D. Garner, occupants of the two houses. Valuable advice and assistance were given freely by Dr. F. C. Houghten, Director of ASHVE Research Laboratory, Pittsburgh, Pa.; Prof. R. E. Lewis, Department of Mechanical Engrg., Duke University; Prof. R. S. King and K. H. Henley, of the Georgia School of Technology. Finally, the excellent cooperation of the members of the Committee on Research of the Atlanta Chapter of ASHVE, H. King McCain, T. T. Tucker, C. Boynton Cole, and F. Warren Clare.

DISCUSSION

W. H. BADGETT, College Station, Tex. (WRITTEN): The authors have, I believe, overlooked one of the most important effects secured from the operation of an attic fan, and seemingly have thought of it largely only as a means of reducing the indoor air temperature, and that of the mass of the building. One of the greatest benefits resulting from the use of attic fans is in the early evening hours through introduction into the house of large volumes of cooler outdoor air in which the occupants may place themselves, thus enjoying the combined effects of lowered air temperature and increased evaporation of body moisture with a resulting lower effective body temperature. I believe that it is important to have a larger number of air changes in the early part of the night with possible reduced air movement for sometime after the occupants have retired in order to secure continued pre-cooling of the house for the following day.

In conclusion No. 7, the authors state that the *gross area* of the discharge openings should not be smaller than the intake grille. What they should have referred to is the *effective areas* of these openings, for *gross areas* can be almost anything, whereas the *effective area* is the controlling factor. Tests at College Station have indicated that more unsatisfactory operation will result from undersized discharge openings than from undersized intake grilles. These tests indicate, too, that where possible it is desirable that the net free area or effective area of the discharge opening be somewhat larger than the inlet. This avoids the possibility of building up a back-pressure with the resulting decrease in operating efficiency and increased noise level.

In the 24-hour test run to compare the two houses without the fan in operation, it is not indicated whether the intake grille of the fan was closed during this test. If the grille remained open, there was in all probability a natural air movement from the rooms of the test house out through the attic which would possibly account for the slightly lower temperature in the test house.

It is possible that both the intake and discharge openings in this installation are small for optimum results. However, since the authors have measured the air movement and converted it to air changes, this will not affect the findings in this paper.

Although there is little that is new presented in this paper, it is a valuable addition to knowledge in the field of attic fan operation because it verifies opinions and facts long held by those who have worked with attic fans. As I understand it, this is the foundation for extensive research in the operation of attic fans, and I believe the authors have made a splendid beginning.

In their outline of future work, I suggest that the following be included in their studies:

1. Resistance to air flow of various types of grilles, particularly expanded metal, wooden egg-crate, automatic intake louvers, automatic exhaust swinging doors, and screen wire when used to cover exhaust openings.
2. Determine if possible the over-all resistance of an average attic fan installation under normal operating conditions.
3. Determine the optimum grille velocity from the standpoint of efficient operation and satisfactory noise level.

J. H. WALKER, Detroit, Mich. (WRITTEN): The research project reported in this paper is the first to be sponsored jointly by a chapter and by the Committee on

Research, and opens up new possibilities in chapter activities. The Atlanta Chapter is certainly to be commended for its initiative and perseverance in conceiving and carrying through this project. There are undoubtedly other lines of research in which chapters might participate because there are many phases of heating and air conditioning which differ according to locality and which, therefore, must be studied locally. This paper shows an enormous amount of data considering the scanty funds which were available. The investigators apparently did a careful job.

If the project can be continued during the summer of 1942 there are two points that deserve attention. The first has to do with air velocities, which apparently have not been taken into account. As in previous discussions of the attic fan, too much significance has been attributed to the cooling effect on the structure and too little importance attached to the creation of air currents at proper locations in the rooms. The purpose of the attic fan is to make people comfortable, not to cool the structure, and to neglect air movement is to neglect one of the basic factors in comfort. The air velocity in the rooms is greatly influenced by the amount of openings of the different windows, and the cooling effect on the human body is correspondingly affected. Future work should study that factor.

The second point concerns the matter of mean radiant temperatures. No measurement of thermal environment is complete without including mean radiant temperature preferably measured with the Vernon globe thermometer. Radiation from walls and ceilings is an important factor in attic fan operation. A comparison of the mean radiant temperatures in the test house and in the control house would be interesting and valuable.

J. A. BISHOP, Dallas, Tex.: The statement was made that people who lived in this house seemed to feel that they were more comfortable with the fan operating even at times when the results of this test indicated that the house was getting warmer, because of the introduction of higher outside air temperature. This situation parallels exactly the experience we have found in Texas. After considerable experience with attic fans it seems to be the general opinion that attic fans are started somewhere from 10 or 11 o'clock in the morning and that is contrary to the opinion of many.

I understand that some of the thermocouples were located on the ceiling of the room. When the outside temperature is warmer than the inside of the house, it is still possible that the attic temperature might be higher than the outside air. Under such conditions the ceiling separating the attic space and the occupied space would be somewhere between the outside and attic temperature. Therefore, the ceiling of the house would act as a panel heater and the use of an attic fan to remove the excess heat from the attic would thereby reduce the surface temperature of the ceiling and reduce the radiation. Have you any figures to substantiate this idea?

It might also be pointed out that the effect of radiant heat from the ceiling is to warm the occupants, and only indirectly the air in the occupied space. Therefore, an occupant of the control room would absorb some of the radiant heat and then dissipate it again by convection or evaporation, whereas an occupant of the room equipped with an attic fan would be relieved of this additional load. This, it seems, would account for the fact that, even with the same air motion and the same interior temperature, the occupant is more comfortable when the heat in the attic is removed by an attic fan.

MR. HINTON: I do not have figures with me so it is impossible for me to give you a definite answer at this time. However, we had about the same thought that you have expressed to the effect that the operation of the fan during the day prevented an increase in attic temperature and reduced the ceiling radiation effect.

E. K. CAMPBELL, Kansas City, Mo.: This idea of air circulation for reducing air temperature was utilized some years ago on a church where the inside volume was much greater. In this case the heat capacity of the material was probably less in proportion and we found that while the fan was operating at night, drawing in air from the outside and using from 10 to 15 air changes per hour, we obtained fairly satisfactory results in cooling the interior of the building by morning. By opening the fan room door so as to utilize the air from the basement and closing the building

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tight we could hold the temperature to a fairly low value during the morning service of the church.

W. A. HINTON (Author's Closure): The suggestions given in the discussions of Messrs. Badgett and Walker are appreciated very much. However, most of these points were included in the original plans of this investigation, but insufficient data were obtained during the Summer of 1941 to warrant any further conclusions in this paper. I wish to state that we were not unmindful of the fact that people install attic fans to cool themselves and not their houses, and we hope that sometime in the future data may be presented that will describe attic fan operation in terms of comfort factors.

Mr. Badgett is entirely correct in his criticism of item No. 7 of the summary. We should have said that the sum of the effective areas of openings from the attic should not be smaller than the intake grille. This was a case of a sentence conveying a false meaning as our results are in no way contradictory to the statement made by him on this point. It would, as Mr. Badgett states, be desirable to have the effective area of discharge openings somewhat larger than the intake grille to the fan.

During the 24-hour test comparing the two houses the fan grille was closed, and it was so stated in the paper under heading of Operating Procedure.

No. 1199

PERFORMANCE OF A HOT-WATER HEATING SYSTEM IN THE RESEARCH HOME

By A. P. Kratz,* M. K. Fahnestock,** W. S. Harris† and R. J. Martin,‡
Urbana, Ill.

On January 2, 1940, the Institute of Boiler and Radiator Manufacturers entered into an agreement for cooperative research with the Engineering Experiment Station at the University of Illinois, and subsequently built, furnished, and completely equipped a house specifically for research work in steam and hot-water heating. It was completed early in January, 1941, and was dedicated as the I=B=R Research Home with appropriate ceremonies on May 23, 1941.

IN the broadest sense the object of these investigations is to study, under actual service conditions, the different types of steam and hot-water heating systems, and various changes therein, from the standpoints of the performance of the systems and their component parts and the environment produced within the building. Corollary studies of the factors related to the performance of the structure are also to be undertaken.

The immediate objects of the tests discussed in this paper were to compare the operating characteristics of a one-pipe, forced-circulation, hot-water heating system under two conditions of operation, and to study some of the factors related to the heat losses from the structure.

Description of Equipment

Research Home: The Research Home, shown in Fig. 1, is a two-story structure, typical of brick veneer on frame construction, and has all of the outside walls and the second floor ceiling insulated with mineral wool bats 3½ in. thick. A vapor barrier, consisting of a glossy surfaced, asphalt impregnated sheathing paper weighing 50 lb per roll of 500 sq ft applied with overlapping joints and as continuous as possible to the inside of the studs of all exterior walls, between the house and the garage, and on the bottom of the second floor ceiling joists, offers resistance to the flow of water vapor from the

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heated spaces into the insulation or to the outdoors. The exterior walls are constructed as shown in Fig. 2 and Table 2. All windows and outside doors are weather stripped, and during the past winter two storm doors were used. Figs. 2 and 3 show the first and second story floor plans, and Table 1 gives a summary of the calculated heat losses for the house. Details of all the types of recesses are shown in Figs. 2 and 3 and Table 3.

In order to make the Home easily adaptable to future test work it was necessary to provide some special structural features, particularly in the details of the plaster construction. Removable panels of sheet rock instead of plaster were used on sections of the walls over all hot water pipes in order to insure accessibility. The radiator recesses were so constructed that radiators with recess panels having inlet and outlet grilles can be used. In order to accom-

FIG. 1. STEAM AND WATER HEATING RESEARCH HOME

modate radiators of different sizes, provision has also been made for changing the size of the recesses with minimum marring of the interior finish.

Heating System: The initial heating plant in the Research Home consisted of a one-pipe, forced-circulation, hot-water system installed in connection with a three section, cast-iron, oil-burning boiler having a net I=B=R rating of 63,000 Btu per hour. A diagram of the arrangement of the heating system and piping layout is shown in Fig. 4. The oil burner was of the pressure atomizing, conversion type, and was fired at a rate of 1.1 gal per hour during the on-periods, with oil weighing 7 lb per gallon having a calorific value of 19,550 Btu per pound. The refractory combustion chamber was 11 in. wide by 12 in. long by 12½ in. high, inside dimensions.

Small tube 19-in., 4-tube, cast-iron radiators were used in open recesses below the windows. The radiators were connected to two supply loops as shown in Fig. 4. The amount of radiation installed in each room was based on a heat emission rate of 200 Btu per sq ft of steam radiation with a mean water temperature of 195 F and a 10 F drop through the radiator, and was

just sufficient to offset the calculated heat loss at an outdoor temperature of −10 F with the 1¼-in. circulator operating continuously.

Instruments: While the house was being constructed, approximately 100 copper-constantan thermocouples made of No. 22 B&S gage wire were permanently installed in the building materials in order to measure temperatures at important points in the structure under various operating conditions. At each of eight locations, one on each exposure of each story, nine thermocouples were installed to provide for establishing complete temperature gradiants

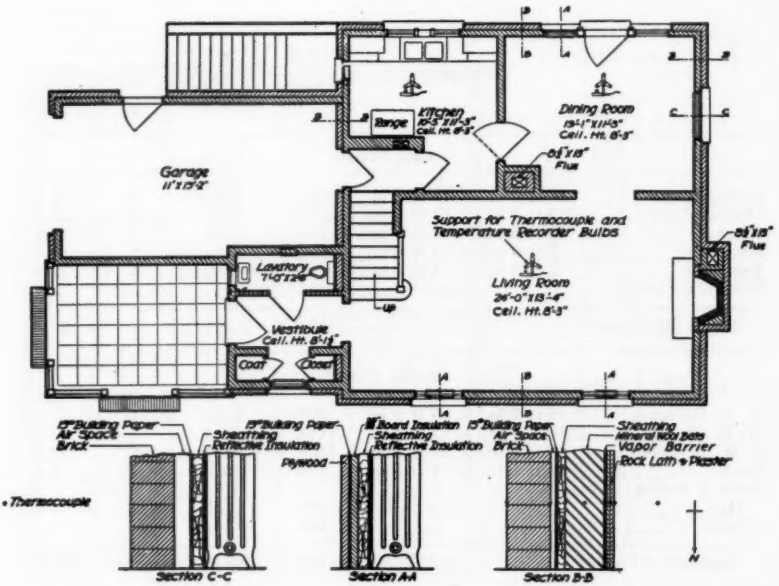

FIG. 2. FIRST STORY PLAN WITH RADIATOR AND INSTRUMENT LOCATIONS AND RADIATOR RECESS DETAILS

through the walls. Other thermocouples were installed in each recess to measure the temperature of the inside and outside surfaces of the sheathing directly back of the radiator. Another group of approximately 50 thermocouples provided for the measurement of air temperatures at various levels in each room, in the attic, and in the basement. A fourth group made it possible to study the performance of the component parts of the heating system. In addition to determining the quantity of water being circulated through each radiator, provision was made for measuring the temperature of the water entering and leaving each radiator, and also the temperature of the water to and from the boiler.

All of the thermocouples were connected to selector switches on a central switchboard, located in the basement, by means of which the electromotive

force of each thermocouple could be quickly read on a precision potentiometer used in connection with a highly sensitive galvanometer. One set of wall temperature gradient thermocouples in each of the four exposures could also be connected to a ten-point recording potentiometer, from which a continuous printed record of the wall temperature gradient over any desired length of time could be obtained.

TABLE 1—DATA ON RESEARCH HOME AND HEATING SYSTEM

ROOM	DIMENSIONS	HEATED SPACE, CUBIC FEET	CALCULATED HEAT LOSS, BTU PER HR WITHOUT STORM SASH	INSTALLED RADIATION*	
				Number of Radiators	Square Feet Steam Radiation
FIRST FLOOR					
Living Room	24 ft 0 in. × 13 ft 4 in.	2641	5749	2	30.8
Dining Room	13 ft 1 in. × 11 ft 3 in.	1183	8742	2	47.6
Kitchen	10 ft 5 in. × 11 ft 3 in.	799	3199	1	19.6
Lavatory	7 ft 0 in. × 2 ft 8 in.	152	1484	1	8.4
Vestibule	7 ft 5 in. × 5 ft 4 in.	284	4848	1	22.4
Vestibule Closet	.....	54	...	...	...
Totals (1st Floor)	.....	5113	24022	7	128.8
SECOND FLOOR					
N. E. Bedroom	10 ft 7 in. × 9 ft 9 in.	800	4393	1	22.4
N. W. Bedroom	10 ft 6 in. × 13 ft 4 in.	1148	4944	1	25.2
S. W. Bedroom	13 ft 0 in. × 11 ft 4 in.	1108	5251	1	28.0
Bath	6 ft 6 in. × 7 ft 6 in.	374	2606	1	14.0
Stair Landing and Hall	.....	505	2155	1	11.2
Closets	.....	345	...	...	...
Totals (2nd Floor)	.....	4284	19349	5	100.8
Totals (1st and Floors)	.....	9397	43371	12	229.6
Basement	24 ft 0 in. × 25 ft 2 in.	5084	29276	Not being heated	

* All 1st and 2nd floor radiators, 19-in., 4-tube, small-tube type. Recessed rating = 1.4 sq ft steam radiation per section.
Mean water temperature 195 F, drop of 10 F, heat emission rate 200 Btu per sq ft per hour.
Net I=B=R Boiler Rating = 63,000 Btu per hour. Oil Nozzle 1.20 GPH.

Moisture measuring stations were installed in approximately 85 strategic locations in the building, conforming with recommendations made by the United States Forest Products Laboratory, Madison, Wis. The measuring stations were located at points in the building at which the moisture content of the materials was most likely to be critical or too high, and were all connected through copper lead wires to a central switchboard located in the basement. By means of the switchboard used in connection with a special mois-

ture meter, the moisture content of the studs, plaster, sheathing, trim, floors, and wall insulation could be quickly observed.

The rate of flow of water in the mains and through each radiator in the heating system could be measured by means of a sensitive differential pressure gage, or manometer, used in connection with sixteen elbow meters.[1] These meters proved highly satisfactory for use in measuring the flow of water in the heating system inasmuch as they introduced no additional resistance to flow. Use is made of the principle that a fluid flowing through an elbow

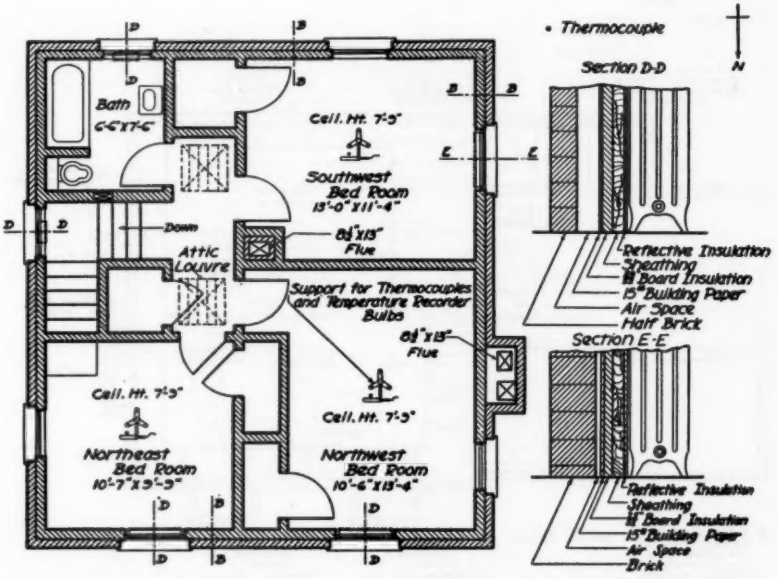

FIG. 3. SECOND STORY PLAN WITH RADIATOR AND INSTRUMENT LOCATIONS AND RADIATOR RECESS DETAILS

creates a difference in pressure between one point on the inside radius and another point on the outside radius of the elbow. This difference in pressure is proportional to the rate of flow.

Recording thermometers were used to make continuous records of the air temperatures in each of the six rooms, the outdoor air, and the flue gas temperatures. The CO_2 in the flue gas was measured by means of both a thermal conductivity type recorder and an Orsat apparatus. The moisture content of the air was measured by means of four humidity indicators, one recording hygrometer, and one wet- and dry-bulb recorder which were checked periodically with an aspirated psychrometer.

[1] The Use of an Elbow in a Pipe Line for Determining the Rate of Flow in the Pipe. University of Illinois, Engineering Experiment Station Bulletin No. 289.

METHODS OF TESTING

During all tests reported in this paper the operation of the heating plant was controlled by a heat anticipating thermostat located 30 in. above the floor in the living room. The average room air temperature at the 30-in. level was maintained constant at approximately 72 F both day and night. At 7:00 A.M., 11:00 A.M., 3:00 P.M., 7:00 P.M., and 11:00 P.M. observations were recorded of the room air temperatures 3 in., 30 in., and 60 in. above the floor and 3 in.

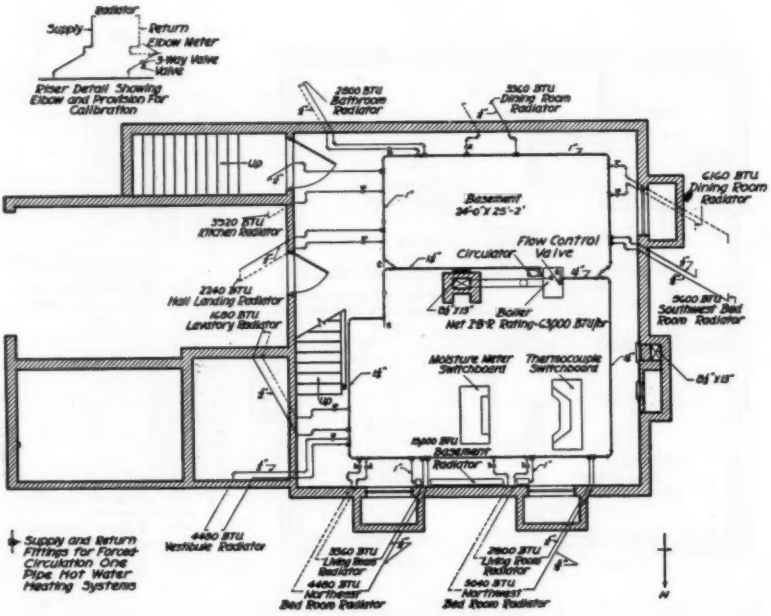

FIG. 4. BASEMENT PLAN WITH LOCATION OF HEATING AND TESTING EQUIPMENT

below the ceiling; the air temperature in the basement, the garage, and the attic; the relative humidity in the heated portion of the house; the temperature inside and outside the sheathing back of each radiator; and of the temperature gradient through the walls of the house. Complete daily records were also made of the operating time, the number of cycles and the power consumption of the oil burner and circulator, and of the weight of oil consumed. Continuous records were made of the stack temperature, the CO_2, the temperature of the water at the boiler outlet and return, the outdoor air temperature, the indoor relative humidity, and the air temperature in each room 3 in. and 30 in. above the floor and 3 in. below the ceiling. Other daily observations included the total amount of electricity and gas used in the

house, the number of occupants, general weather conditions, and moisture content of various parts of the house structure such as sheathing, studs, floor joists, plaster, and finish trim.

Two series of tests were run which differed only in the operating sequence of the oil burner and circulator. In the first series, designated as series *A*, the control arrangement was that shown in Fig. 5. In series *B* the low limit aquastat was eliminated, and the flow control valve was locked open so that it could not prevent the circulation of water in the heating system at any time. With this method of control the radiators could continue to cool all of the water in the system as long as sufficient temperature difference persisted to

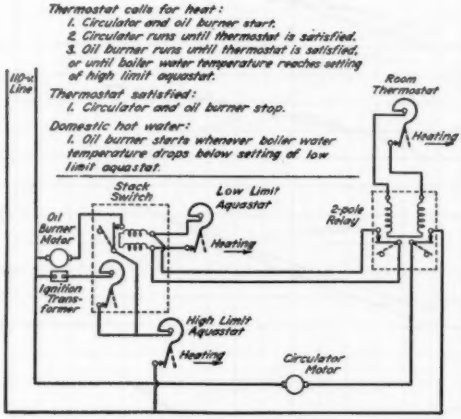
Thermostat calls for heat:
1. Circulator and oil burner start.
2. Circulator runs until thermostat is satisfied.
3. Oil burner runs until thermostat is satisfied, or until boiler water temperature reaches setting of high limit aquastat.
Thermostat satisfied:
1. Circulator and oil burner stop.
Domestic hot water:
1. Oil burner starts whenever boiler water temperature drops below setting of low limit aquastat.
115-v. Line
Room Thermostat
Heating
Stack Switch
Low Limit Aquastat
2-pole Relay
Oil Burner Motor
Heating
Ignition Transformer
High Limit Aquastat
Heating
Circulator Motor

FIG. 5. INITIAL CONTROL SYSTEM IN RESEARCH HOME

effect any circulation by gravity, and the temperature of the water in the boiler could vary anywhere between that of the room and 200 F.

RESULTS OF TESTS

Performance of Plant: A comparison of the burner performance curves for series *A* and *B*, shown in Fig. 6, indicates that in mild weather, or at low indoor-outdoor temperature differences, operation on series *A*, with the low limit aquastat and the flow control valve operative, resulted in increased burner operating time, burner power, and fuel consumption. These increases were caused by the larger heat losses from the chimney gases accompanying the higher average boiler water temperature maintained in mild weather in series *A* as compared with that maintained in series *B*, for which the low limit aquastat was inoperative and the flow control valve was open all of the time. As the indoor-outdoor temperature difference became larger, the average boiler water temperature for series *B* automatically approached that for

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series *A* as a result of the more frequent burner operation required to maintain the normal room temperature of 72 F. At approximately 55 F indoor-outdoor temperature difference, the curves for series *A* became tangent to those for series *B*, and at all temperature differences above this the operating characteristics for both series were the same, thus indicating that for indoor-

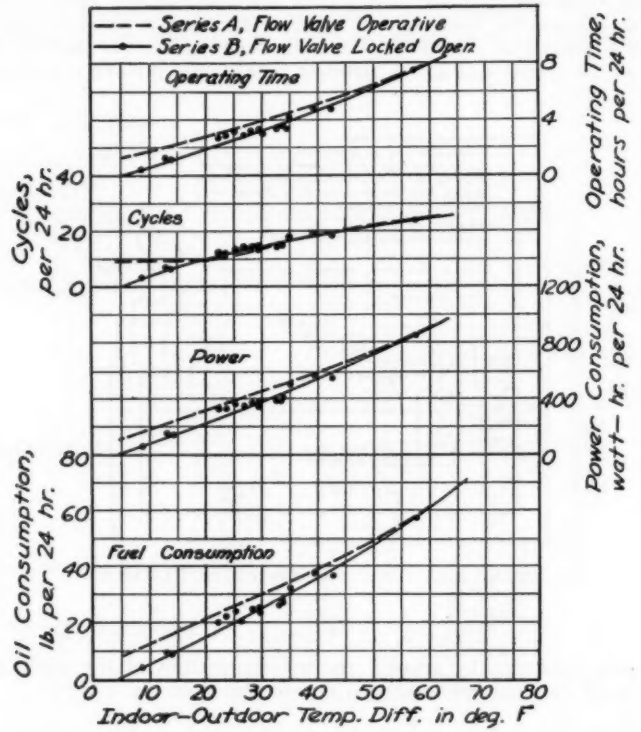
Series A, Flow Valve Operative
Series B, Flow Valve Locked Open
Operating Time
Cycles
Power
Fuel Consumption
Cycles, per 24 hr.
Operating Time, hours per 24 hr.
Power Consumption, watt-hr. per 24 hr.
Oil Consumption, lb. per 24 hr.
Indoor-Outdoor Temp. Diff. in deg. F

FIG. 6. BURNER PERFORMANCE CURVES, SERIES *A* AND *B*

outdoor temperature differences above 55 F the amount of burner operation required to maintain normal room temperature was sufficient to maintain a minimum water temperature of 165 F in the boiler during the off-periods of the burner. In milder weather the room thermostat did not operate the burner over a long enough period to maintain the boiler water at 165 F, and as a result, in series *A* the low limit aquastat continued the operation of the burner for a short time at the end of each cycle after the room temperature became normal and the circulator ceased to operate. The excess in the amount of fuel burned, power consumed, and time of burner operation shown for

series *A* over that shown for series *B* is the amount of each required to maintain sufficient temperature in the boiler to make it possible to use the latter as a means of heating domestic hot water. Since the internal water heater was not connected to the hot water storage tank during these tests, the results do not include any data on the standby loss from the storage tank, or on the requirements for the actual heating of water for domestic uses.

In Urbana, Ill., the average outdoor temperature during the heating season is approximately 38 F. Hence, if the house temperature is maintained at 72 F the average heating day is one in which the indoor-outdoor temperature difference is 34 F. The average length of the heating season is 204 days.

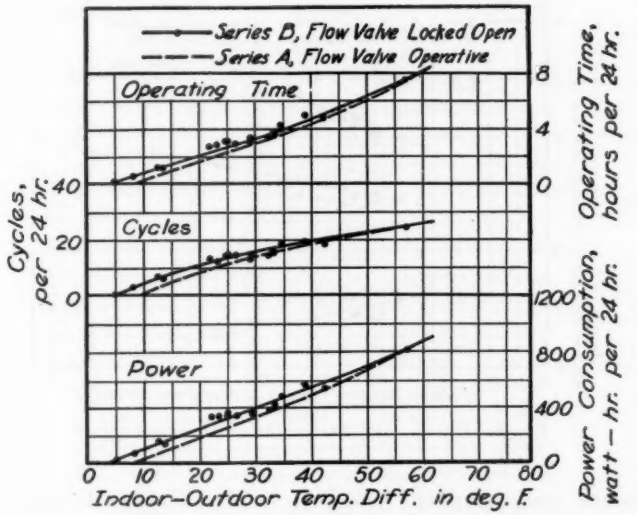

FIG. 7. CIRCULATOR PERFORMANCE CURVES, SERIES *A* AND *B*

Comparisons based on this temperature difference may be regarded as indicative of the results to be secured from the entire heating season. From Fig. 6 it is evident that 33.5 and 30 lb of oil per 24 hours were required in series *A* and *B*, respectively, at an indoor-outdoor temperature difference of 34 F, or since the fuel oil used weighed 7 lb per gallon, operation on series *A* required an additional ½ gal of oil per average heating day or 102 gal per heating season over that required on series *B*.

No significant difference existed in the number of burner operating cycles required for series *A* and *B* except in very mild weather, in which case a minimum of 9 cycles per day was required in series *A* in order to maintain a minimum boiler water temperature of 165 F.

Just enough radiation was installed in the house to offset the calculated heat loss at −10 F, when the radiators were constantly supplied with water entering and leaving at 200 F and 190 F respectively. This would require

continuous circulator operation. By extrapolating the circulator operating time curve, shown in Fig. 7, to 80 F temperature difference, a total operating time of only 12 hours per day is obtained instead of 24. Superficially this would seem to indicate that the radiators were considerably oversize for the house. However, sufficient data have not been obtained at present either to confirm or to refute this. Further data will be obtained on this in future studies relating to over-all house efficiency, or utilization of heat in the house.

At a given indoor-outdoor temperature difference, the circulator operating time, power, and number of cycles were slightly less in series *A* than in series *B*. This condition resulted from the fact that higher average radiator

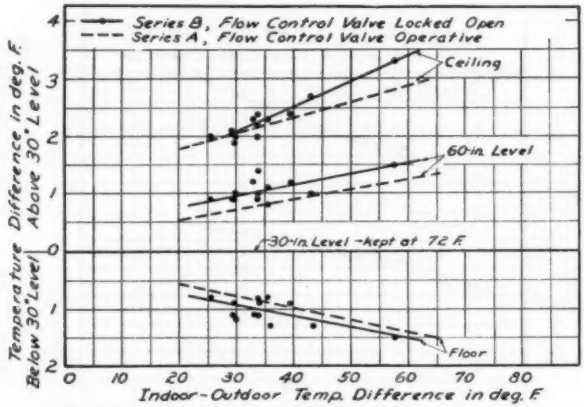

FIG. 8. INDOOR AIR TEMPERATURE DIFFERENTIALS

temperatures, and consequently higher rates of heat input to the rooms, were obtained during the on-periods of the circulator in series *A* than in series *B*.

Room Temperature Gradients: The maximum difference in the average temperature of the air at the 30-in. level observed in any two rooms of the house was 2.8 F. The maximum differences observed on the first and second stories separately were 1.2 F and 1.5 F respectively. The second story temperature averaged about 1.3 F higher than that for the first story.

With the indoor air temperature at the 30-in. level kept constant at 72 F, the air temperature difference from floor to ceiling, shown in Fig. 8, increased with decreasing outdoor air temperatures, and the method of plant operation had no material effect upon this gradient. In the coldest weather experienced there was a difference of only approximately 4.5 F between the average temperature of the air 3 in. above the floor and that 3 in. below the ceiling. That the elimination of the action of the flow control valve in series *B* caused no over-runs in the air temperatures within the house is indicated by the fact that a difference of only 0.5 F between the maximum and minimum air temperatures at the 30-in. level was obtained during a cycle of operation. The maximum variation at any level during a cycle was only 0.9 F, and this

occurred near the ceiling. These values are all well within the limits of control to be expected from the average room thermostat available today.

Wall Temperature Gradients: By means of thermocouples located in the walls, as shown in section "B-B," Fig. 2, it was possible to obtain the temperature gradient through any wall at any given time. Fig. 9 shows a typical temperature gradient through the north wall on a day when the outdoor temperature averaged 10 F. On the assumption that a condition of steady flow of heat through the wall exists in this case, an assumption always made in the

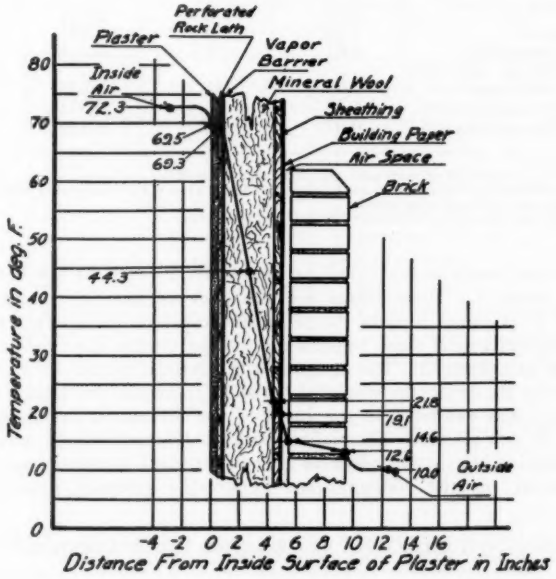

FIG. 9. TYPICAL TEMPERATURE GRADIENT THROUGH NORTH WALL OF RESEARCH HOME

calculation of heat losses from buildings, Fig. 9 offers a means of comparing the actual insulating value of any of the building materials used in the wall with the theoretical value. If the observed temperature drops are accepted as being representative of the actual conductances of the corresponding constructions, and the drop through the 3⅝ in. combined mineral wool and studs is used as a base, then the insulating values computed from the ratios of the different temperature drops to the base drop may be regarded as actual insulating values. The insulating values computed from the ratios of the commonly accepted values of the conductances may be regarded as theoretical values. Table 2 shows equivalent insulating values computed by both methods. Column 4 expresses the theoretical thickness in inches of an hypothetical characteristic insulation having the same conductivity as that of the combined mineral

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TABLE 2. EQUIVALENT INSULATION VALUES OF MATERIALS AS USED IN WALLS IN RESEARCH HOME

ARRANGEMENT OF MATERIALS AS USED IN WALLS	CONDUCTANCE FOR CONSTRUCTION STATED IN COL. 1 C	OBSERVED TEMP. DROPS THROUGH CONSTRUCTION DEG F ΔT	THICKNESS OF CHARACTERISTIC INSULATION[b] EQUIVALENT TO CONSTRUCTION IN COL. 1 IN INCHES	
			Based on C	Based on ΔT
1	2	3	4	5
4-in. brick	2.30	2.0	0.141	0.153
½-in. air space[c]	1.10	4.5	0.295	0.345
25/32-in. wood sheathing	1.02	2.7	0.318	0.207
3 5/8-in. combined mineral wool and wood studs	0.089[a]	47.3	3.625	3.625
¾-in. rock lath and plaster	4.40	0.4	0.074	0.031
9.66-in. wall as constructed	0.079	56.9	4.114	4.360

[a] Conductivity of combined mineral wool and wood studs = 0.325.
[b] Characteristic insulation assumed as one having a conductivity of 0.325 conforming with conductivity of combined mineral wool and wood studs.
[c] Not included in computed conductance for wall as constructed.

wool and wood studs which would be equivalent to each of the constructions listed in Column 1. These values are based on conductance values, C, given in Column 2. Column 5 gives the actual thickness of the characteristic insulation equivalent to the same constructions based on the observed temperature drops given in Column 3. The actual and theoretical thicknesses were in close agreement for the brick, the insulated studding space and the total wall. Some disagreement was exhibited in the cases of the air space, the sheathing, and the rock lath and plaster. These discrepancies can probably be accounted for by deviations in the actual physical dimensions and moisture contents from those assumed for the computations. Following common practice, no air

TABLE 3—NORMAL WALL AND RADIATOR RECESS CONSTRUCTION

NORMAL WALL SECTION	RADIATOR RECESS SECTIONS		
	S. W. Bed Room	Dining Room	Living Room
Lath and Plaster			
Vapor Barrier[a]			
Mineral Wool Insulation			
	Reflective Insulation	Reflective Insulation	Reflective Insulation
25/32-in. Wood Sheathing	25/32-in. Wood Sheathing	25/32-in. Wood Sheathing	25/32-in. Wood Sheathing
	25/32-in. Board Insulation		25/32-in. Board Insulation
15 lb Felt building paper	15 lb Felt building paper	15 lb Felt building paper	15 lb Felt building paper
½-in. Air Space	½-in. Air Space	1 in. Air Space	
Full Brick	Full Brick	Full Brick	¼-in. Plywood

[a] Glossy surfaced, asphalt impregnated sheathing paper, 50 lb per roll of 500 sq ft.

space between the brick and sheathing was allowed for in the computation of the over-all conductance for the wall. However, the data indicate that in the actual wall sufficient insulating value was derived from the space between the brick and the sheathing to offset the deficiencies shown in the cases of the sheathing and the lath and plaster, so that the actual and theoretical insulating values for the total wall were in close agreement.

As shown by Fig. 9, the temperature of the inside surface of the exposed wall was only 2.8 F lower than the room air temperature. This difference remained practically constant, irrespective of the changes in the temperature outdoors and in that of the outer layers of the wall. From the standpoint of

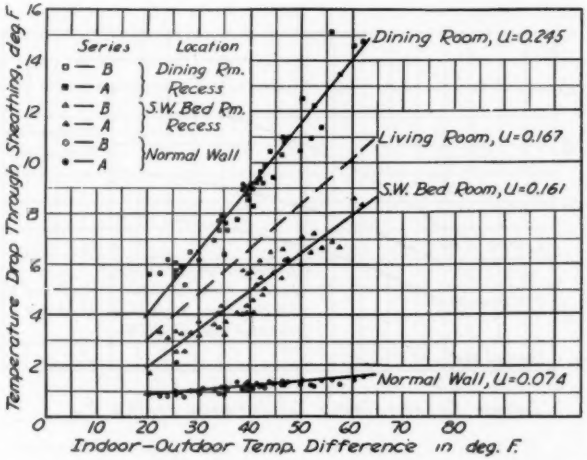
Temperature Drop Through Sheathing, deg F
Indoor–Outdoor Temp. Difference in deg. F.
Series
Location
B — Dining Rm. Recess
A — Dining Rm. Recess
B — S.W. Bed Rm. Recess
A — S.W. Bed Rm. Recess
B — Normal Wall
A — Normal Wall
Dining Room, U=0.245
Living Room, U=0.167
S.W. Bed Room, U=0.161
Normal Wall, U=0.074

FIG. 10. TEMPERATURE DROP THROUGH SHEATHING FOR THREE DIFFERENT RECESS CONSTRUCTIONS AND NORMAL WALL

comfort the high temperature of the inside surface of the exposed walls is of considerable importance in that it reduces the amount of heat lost by the human body by radiation. Furthermore, it tends to minimize the floor to ceiling temperature difference, thus minimizing the tendency for cold floors brought about by downward currents of cold air chilled by the presence of cold wall surfaces.

Heat Transmission through Radiator Recesses: Throughout the heating season the temperatures of the inside and outside surfaces of the sheathing directly back of each radiator were observed. The records for three of the recesses are of interest, in that they show how the construction of the recesses may affect the heat loss from the house. Table 3 gives details of the construction of the cross section of each of the three recesses and of the normal wall. In Fig. 10 the observed temperature drops through the sheathing are plotted against the indoor-outdoor temperature differences for a normal wall section and for each of the three recesses. The method of operating the heating

plant had no apparent effect on these temperature drops. Since the conductance of the sheathing is the same in each case, the ratio of the temperature drops through the sheathing for any two sections is also representative of the ratio of the actual heat transmission through the sections. Therefore, for any given indoor-outdoor temperature difference, if the temperature drop through the sheathing for any given recess is divided by the temperature drop through the sheathing for the normal wall, a value which may be designated as the heat transmission ratio for that recess is obtained. This figure is also equivalent to the square feet of normal wall required to transmit the same quantity of heat per hour as that transmitted by one square foot of recess area. These heat transmission ratios have been calculated for each recess and

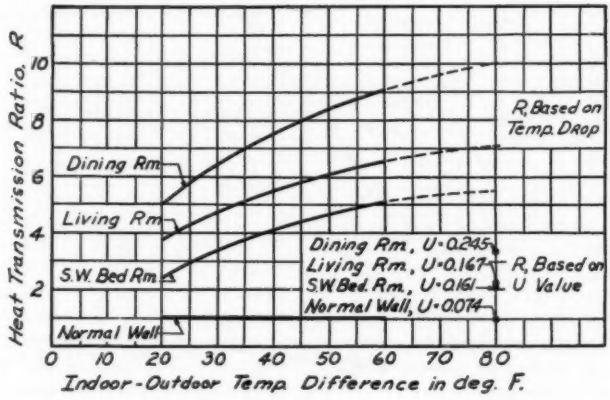

FIG. 11 HEAT TRANSMISSION THROUGH THREE DIFFERENT RECESS CONSTRUCTIONS AS COMPARED WITH NORMAL WALL

are shown plotted in Fig. 11. The isolated points represent the ratio of the calculated over-all heat transmission coefficient for each recess to that for the normal wall. The discrepancies between these values and values read from the curves at given indoor-outdoor temperature differences result from the fact that the temperature back of the radiators was higher than that of the air in the room. The increase in the heat transmission ratios as the indoor-outdoor temperature difference increased was undoubtedly brought about by the fact that higher average radiator temperatures required as the weather became colder resulted in greater differences between the temperature back of the radiator and that in the room.

In interpreting these curves the values at two indoor-outdoor temperature differences should be considered. The first is at an indoor-outdoor temperature difference of 80 F, since this represents design temperature, and hence values at this difference are related to the sizing of the heating system. The second is at an indoor-outdoor temperature difference of 34 F since this represents average winter weather in Urbana, Ill., and hence, has a bearing upon the seasonal operating costs.

No −10 F weather occurred during the testing season, but by extrapolating the curves in Fig. 11 a heat transmission ratio, R, of approximately 10 may be obtained for the uninsulated dining room recess. That is, one square foot of area in this recess, with a U value of 0.245, would transmit the same quantity of heat per hour as 10 sq ft of normal wall with a U value of 0.074. There were 53.25 sq ft of recess area in the house, for which no allowance was made in computing the design heat loss. If every recess in the house had been of the same construction as the one in the dining room, an additional heat loss of $(10 - 1) \times 0.074 \times 80 \times 53.25 = 2837$ Btu per hour would have resulted, or an increase of 6.55 per cent in the calculated heat loss for the house without storm sash. This is equivalent to 10 sections of the radiation used.

Similarly, using the heat transmission ratio and the calculated heat loss at 34 F indoor-outdoor temperature difference, it may be shown that if the rigid insulation had been omitted from all of the recesses, the seasonal fuel consumption would have been increased by about 4.4 per cent over that to be expected with the use of free standing radiators.

The S. W. bedroom recess, section E-E, Fig. 3, was of the same construction as the one in the dining room except for the addition of 1 in. of rigid insulation between the brick and sheathing. The addition of this insulation changed the calculated U value from 0.245 to 0.161. Likewise the heat transmission ratio was reduced from 10 to 5.5 at 80 F indoor-outdoor temperature difference and from 6.9 to 3.7 at a temperature difference of 34 F. Thus the addition of the 1 in. of rigid insulation reduced the heat loss through the recess by

$$\frac{10 - 5.5}{10} \times 100 = 45 \text{ per cent}$$

at an indoor-outdoor temperature difference of 80 F and by

$$\frac{6.9 - 3.7}{6.9} \times 100 = 46.4 \text{ per cent}$$

at 34 F temperature difference. It is true that the actual magnitudes of the total heat losses through these recesses are small, but they do represent from 2 to 4 per cent increase in the seasonal fuel bill, and these charges continue year after year. Also, the poorest recess included in this study is comparable with what is commonly regarded as fairly high quality construction. With poorer recess construction the heat losses could easily become of such a magnitude as to seriously affect both the operating cost and the size of the heating plant required for a given installation.

Operating Costs: The curves in Fig. 12 give the daily operating costs obtained over the range of outdoor temperatures experienced during the testing season. These curves were obtained by multiplying the amount of fuel and power used at various indoor-outdoor temperature differences, as shown on the burner and circulator performance curves in Figs. 6 and 7, by the three unit costs for oil and power, shown in Fig. 12, thus determining the daily cost of operation. Of the three sets of unit costs selected, the 7¼ cents per gallon for oil and the 3½ cents per kilowatt hour for power were those

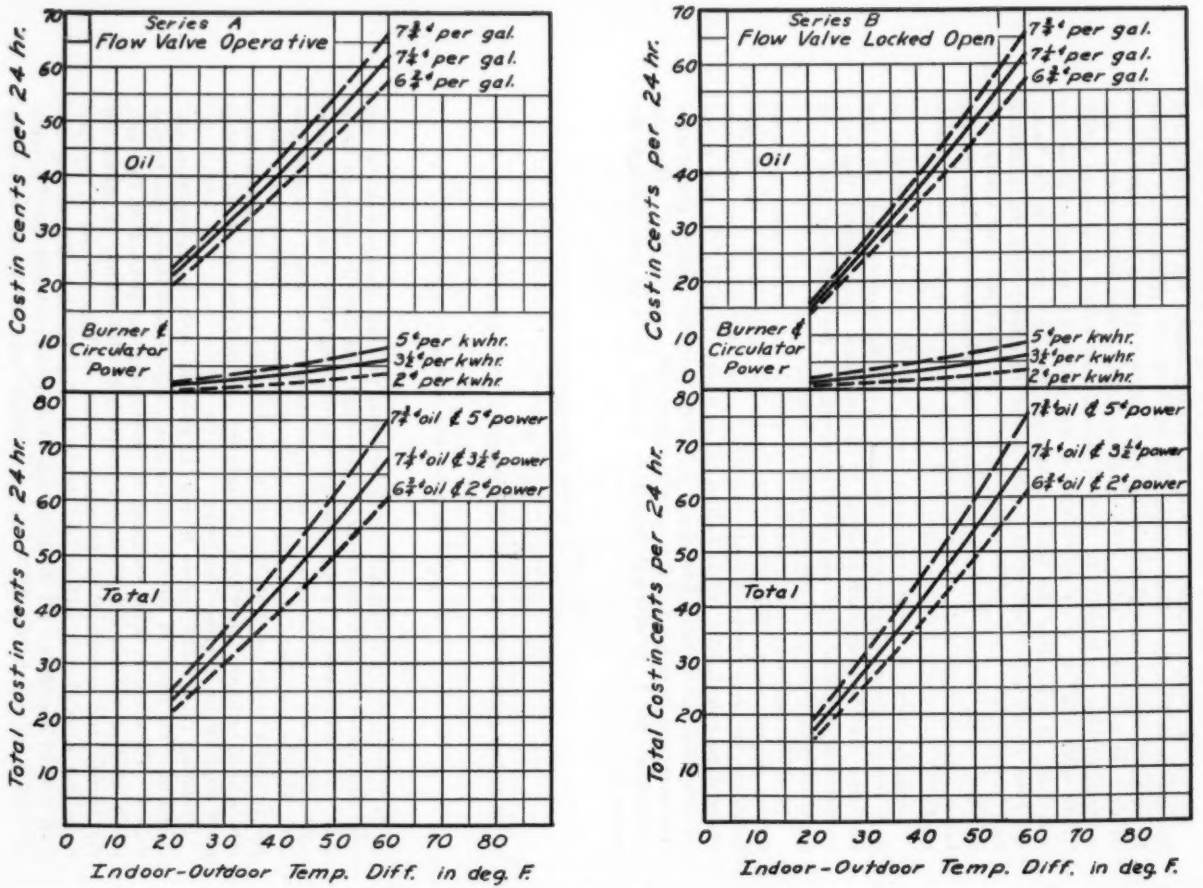

FIG. 12. OPERATING COSTS, SERIES *A* AND *B*

prevailing in Urbana, and the total cost curves calculated from these rates were used in the following seasonal cost calculation. The average heating season in Urbana consists of 204 days at an average temperature of 38 F or 5500 degree-days. Thus, with an indoor temperature of 72 F, the indoor-outdoor temperature difference on the average heating day is 34 F. The average daily total cost from the curves would be \$0.38 for series *A* and \$0.34 for series *B*. The total seasonal cost of operation would therefore be $204 \times 0.38 = \$77.50$ for series *A*, and $204 \times 0.34 = \$69.40$ for series *B*. The difference, \$8.10, represents additional stack losses from the boiler resulting from the maintenance of a minimum boiler water temperature of 165 F. This would be required if the boiler were used to heat domestic hot water, and this cost of \$8.10 would be in addition to the cost of the fuel required to heat any domestic water used.

SUMMARY

The following is a summary of the results obtained in this investigation:

1. Operation of the plant with the low limit aquastat maintaining a minimum boiler water temperature of 165 F and with the flow control valve preventing circulation of water during the off-periods of the circulator resulted in an increase in the seasonal operating cost of approximately 11.7 per cent as compared with operation with the low limit aquastat eliminated and the flow control valve locked open. The increase in fuel consumption was caused by increased heat losses from the top of the chimney accompanying the higher average boiler water temperature maintained in mild weather with the flow control valve effective.

2. No over runs in the temperature of the air in the rooms were experienced with either method of operation. The maximum variation in air temperature at the 30-in. level in any room did not exceed 1 deg F.

3. In severe weather the average difference between the temperature of the air 3 in. below the ceiling and that 3 in. above the floor was of the order of 4.5 F. The method of operating the heating system had no material effect upon this temperature difference.

4. The temperature of the inside surfaces of the exposed walls in severe weather was only about 2.8 F lower than that of the air in the rooms.

5. The application of one inch of rigid insulation to a radiator recess lined with reflective insulation, backed by wood sheathing, one inch air space and 4-in. brick veneer resulted in a reduction of approximately 45 per cent in the heat loss as compared with that lost through the same recess without insulation.

ACKNOWLEDGMENTS

The results presented in this paper were obtained in connection with the investigation of steam and hot water heating systems in the Research Home at the University of Illinois, conducted by the Engineering Experiment Station of which M. L. Enger, Dean of the College of Engineering, is the director, and in the Department of Mechanical Engineering, of which O. A. Leutwiler, Professor of Mechanical Engineering Design, is the head. This investigation is a cooperative project sponsored jointly by the *Institute of Boiler and Radiator Manufacturers* and the Engineering Experiment Station. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is hereby made to the various manufacturers who cooperated by furnishing materials and equipment.

Indoor-Outdoor Temp. Diff. in deg. F.

Indoor-Outdoor Temp. Diff. in deg. F.

FIG. 12. OPERATING COSTS, SERIES *A* AND *B*

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DISCUSSION

J. C. FITTS, New York, N. Y. (WRITTEN): This paper, while it deals with a type of system with which all of us are familiar, is of fundamental importance because in it we are dealing with integrated results. In the past we have established a number of interesting, important, separate facts and used them as the foundation of our method of design. We have tested radiators and boilers to determine what they would do in a heating system. We have found the friction losses in pipes. We have determined the heat transmission through various types of wall construction. We know a great deal about the various parts of a heating system and the conditions we are liable to meet in different kinds of buildings.

Now our research is taking a fundamental departure from the study of parts and we are to have a careful study of the over-all operation of a radiator heating plant under actual service conditions, yet with facilities for accurate observation and scientific control. As a check upon our practice this should prove very valuable.

An interesting feature of the report is the demonstration of our ability to distribute heat evenly throughout a home. The small temperature differences within the rooms and between the various rooms confirms the validity of our methods of design. This would lead one to the belief that we have enough knowledge of isolated facts to do our job well and that now our research should be directed toward the solution of problems arising from the study of complete systems in complete structures which after all is the environment in which the human being has his existence. Equally important is the study of the reaction of the individual and of groups of individuals to this environment which we help to create.

These studies may uncover the need for further research in connection with parts of the system but it will then remain subordinate to the main research projects. It is to be hoped that this work will be continued.

A. G. DIXON, Racine, Wis. (WRITTEN): Under Results of Tests, comparison of series *A* and series *B* oil burner control method brings out distinct economy advantage (approximating 10 per cent in fuel plus power costs). For series *B* method; this is to be expected, since series *B* control procedure results in low boiler water temperature when outdoor temperatures are mild. All tests reported omit the use of heating system for provision of domestic hot water and series *B* control method would not be feasible if domestic hot water were being provided.

It is presumed that future tests will incorporate employment of facilities for generating supply of domestic hot water within the residence heating plant. It is well known that in many parts of the country a higher cost of fuel must apply if domestic hot water is heated separately from the heating plant. Even where this is not the case, stand-by losses are inevitable in separate domestic hot water heaters. Either or both of these factors might operate to negate the fuel economy differential now apparently favoring series *B* method.

While the Research Home cannot deal in the scores of possible variations of fuel and equipment costs, that might apply with various systems and in various parts of the country, further tests to include integral operation of domestic hot water heating could apparently be quite profitable.

Most interesting single unanswered question in the report is that radiation installed in the Research Home is calculated to be only enough to heat the house when radiation is continuously provided with water averaging 195 deg temperature in the radiator. Yet under (extrapolated) extreme condition of —10 F outdoor temperature, circulator operated only 12 hours per day instead of 24. It is noted that superficially this would seem to indicate that radiators are oversized for the house.

If we are prompted by this to assume that standard methods of heat loss calculations now indorsed by the Society result regularly in the provision of an excess of radiation, then we may well inquire, with particular reference to residence application,

is this good or bad? We should not be too quick to say that it is good since it gives a factor of safety. Perhaps it is bad, in view of the tremendously increased use of automatically-fired heating systems in residences during recent years. The best general answer to the problem that we commonly term *cold seventy* discomfort, is a closer approach to continuous operation of the system, *by one or another means.* Like the cook who with good intentions doubles the amount of butter and sugar called for by the cake recipe and thereby gets a bad cake, the engineer who puts a high factor of safety into his radiation calculations may conceivably be hurting performance, rather than helping it. It seems to me that further study of this particular vein of exploration holds exceedingly interesting possibilities.

It is elementary but nevertheless worth noting that the *proportion* of loss of heat from (well insulated versus poorly insulated) recesses, compared with total loss of heat from a house, will vary according to the total character of insulation of the structure. In the Research Home excellent insulation of the structure as a whole is provided; hence any weak spot in the insulating armor of the recesses appears as its worst by comparison. In the light of this arithmetical ratio, it is a fortunate natural tendency to insulate radiator and convector recesses well, in residences that are being generally well insulated. Nothing in this comment, of course, is intended to oppose the logic of advisability of insulating radiator recesses.

More involved is the question that can be raised as to whether identical radiators to those covered by report, but *not recessed,* but rather set out in the room proper in the manner commonly called *exposed,* would give identical room temperature gradients to those experienced when the radiators are recessed, and would otherwise demand only commensurate supply of heating medium in order to satisfy the same comfort level setting of thermostat that was provided in operation where the radiators were recessed. It seems quite likely that the chimney action of recessing of the radiators will have affected the ratio of convected versus radiant heat emanating from these radiators and that this in turn could influence performance results.

What I am trying to say is that tests reported up-to-date do not permit the conclusion that recessing of radiators in any except the best-insulated recesses will, per se, increase total fuel costs. At least it may be questioned that such fuel cost increase is subject to simple calculation based upon wall structure remaining beyond a radiator recess on the one hand, and complete wall structure behind an exposed radiator on the other hand. In short, it seems possible and even likely that another variable, one of performance of radiators according to location in recesses or exposed, applies. Perhaps those who direct its program will be able to justify future Research Home fuel cost investigation, of a kind that will throw further light on this interesting detail.

H. F. HUTZEL, Detroit, Mich. (WRITTEN): The authors indicate in Figs. 6 and 7 that by extrapolating the circular operating time curve, a total operating time of only 12 hours per day is obtained instead of 24. This is particularly interesting in view of the fact that the installed radiation was selected on the basis of being just sufficient to off-set the calculated heat loss at an outdoor temperature of —10 deg, and with the circulator operating continuously.

From data obtained during the test they were not able to offer an explanation for this short operating time. I am wondering if this condition could not in part be accounted for by the fact that the radiation installed was selected on a basis of 200 Btu per square foot per hour. With 195 deg entering water and 185 deg leaving water in a room of 70 deg, the coefficient of heat transmission figures out to be 1.65 Btu per degree temperature difference per hour. As I recall a 3-column large tube radiator 38 in. high has a coefficient of 1.65 under such conditions of operation. I would expect that a 19 in. high radiator, 4 tubes deep and of a small tube type, would have a considerably higher coefficient of heat transfer due to its lower height, and the fact that it possesses less inter-reflecting surface. I would except such a radiator, under stipulated conditions of operation, to have a coefficient of heat transfer nearer 1.85 or approximately 12 per cent greater than that on a basis of which these

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radiators were selected. If this is correct, the installed radiator capacity would represent 51,500 Btu, which is approximately 18½ per cent in excess of the calculated heat loss.

In addition to this increased heat capacity of the radiators, we must also add a percentage of the heat lost by the piping, a large percentage of which is available for compensating for part of the calculated heat loss.

There may be one or two other influencing factors which may help to explain the short running time when operating under design conditions. They are—wind velocity at time tests were conducted, as well as, whether or not the sun was shining continuously during the day time.

I note that the burner used was fired at 1.18 gal per hour during the *on* period. Assuming that the oil used had a heat value of 140,000 Btu per gallon and assuming that the boiler and burner efficiency was 75 per cent, this would provide for a heat output at the nozzle of 115,000 Btu. This capacity is 256 per cent of the calculated heat load. One would expect that a burner of this capacity on the load in question, would short cycle and operate for a short period at a time. I cannot reconcile, however, why the number of cycles for the burner and the circulator should be so closely in accord. If these curves are extrapolated to 80 deg temperature difference they indicate that both cycles of the burner and of the circulator would be approximately 29 cycles in 24 hours. Here again, I am surprised at the curvature of the circulator cycles. If the radiators were properly selected for the load in question one would expect the circulator to operate continuously at 80 deg temperature difference unless the water supply to the radiators should exceed 185 deg. In the case of the burner cycle curves we naturally would expect fewer cycles under full load conditions. The number of cycles, of course, would depend upon the relationship between the capacity of the burner and capacity of the radiation.

I realize that the immediate object of the tests discussed in this paper was to compare the operating characteristics under two different methods of control. This probably explains why the authors did not show a table showing rate of water flow through each radiator, temperature in and temperature out, and the temperature of water in the boiler at the cut-out point. I am wondering if there was not sufficient over rise in temperature to account for the shape of the operating performance curves referred to.

C. J. STERMER, Chicago, Ill. (WRITTEN): The writer has observed one part of the data which might cause a reader to jump to the conclusion it would be bad economics to heat the water for domestic supply by indirect means from the house heating boiler. This does not seem to be borne out in practice. Item No. 1 in the summary lists an 11.7 per cent increase in seasonal operating cost due to the heat losses at the top of the chimney. This might be termed, *stand-by loss*. In figuring the cost of domestic water heating, account would have to be taken of the *stand-by loss*, which would occur if a separate water heater were employed. The *stand-by loss* from the separate water heater, coupled with the probability of a higher cost fuel, could account for a fuel cost that would offset the losses observed in the tests reported and might prove the indirect water heating method the least costly. Also having automatic firing devices operate throughout the year for the water heating may prove desirable from a service standpoint, rather than to have the equipment idle during the summer months.

Another point to consider is that if a separate heater were used during the winter time, there would be a stand-by loss occurring from two devices, the separate water heater and the central heating plant, instead of from one.

It is suggested that in subsequent studies of water heating costs, the stand-by loss from various heaters and effects of idleness on equipment be taken into account in cost of operation comparisons.

F. E. GIESECKE, College Station, Tex. (WRITTEN): The authors state that if the circulator operating time curve of Fig. 7 were extrapolated to 80 F, a total operating

time of only 12 hours, instead of 24 hours, is indicated and that this might indicate that the radiation is considerably oversize for the house.

If, in the same figure, the cycle curve were extrapolated to 80 F, it would indicate 30 cycles in 24 hours. Consequently, every heating cycle would consist on an average of 24 min of forced circulation and 24 min of gravity circulation. The radiators would dissipate heat during the 24-min period of gravity circulation as well as during the 24-min period of forced circulation, but at a lower rate since the average water temperature in the radiator would be lower. If both periods of heat dissipation are taken into account it may show that the radiation is not materially, if any, oversize for the building.

It would be very valuable to have a continuous record of the temperatures of the water leaving and returning to the boiler and also a continuous record of the volume of water flowing through the boiler, determined by means of an elbow meter in the main flow or return line.

From Fig. 12 B it appears that the cost of burner and circulator power is about 10 per cent of the cost of the fuel. It would be valuable to know what portion of this power cost applies to the circulator in order to make a study of the relative economy of a forced circulation system using circulators and an ordinary gravity circulation system.

Finally, the authors state that the average heating season in Urbana consists of 204 days at an average temperature of 38 F or of 5,500 degree-days.

Since the average indoor-outdoor temperature difference is to be 72 minus 38 or 34 F, would it not be well, at this time to propose a new unit and say that the average heating season in Urbana consists of 204 days at an average indoor-outdoor temperature difference of 34 deg or of 6936 degree-days, based on an indoor temperature of 72 F instead of basing it on an indoor temperature of 65 F which is only indirectly related to the heat requirements of the building.

FERDINAND JEHLE, Indianapolis, Ind. (WRITTEN): Items 1 and 2 of the summary are of particular interest. It was rather a surprise that the tests under series *A* did not show some over-run in room temperature. The greater cost of heating during series *A* is of particular interest. It touches on a subject upon which there seems to be very little published data. It is hoped that Professor Kratz will, at some future time, publish a chart showing the variation of stack losses against boiler water temperature.

Some mention is made of radiator supply water temperature but complete data are not included in the paper, as to how this varied against time on both the series *A* and series *B* tests. In mentioning this to one of the authors, he stated that much of these data had to be eliminated from the paper in the interest of space. Original data, such as these, are always of value because they make the work applicable to other problems which members may be confronted with. It is, therefore, suggested that the Society permit the publication of such valuable information, although it might occupy considerable space.

The statement is made in the paper that the maximum difference in the average temperature at the 30 in. level, observed in any two rooms, was 2.8 F. It is not clear just what was averaged but, presumably, the temperature over 24 hours. It would be of interest to know what the maximum variation was between two rooms at any one time.

MR. HARRIS: It was interesting to find that a great deal of the discussion centered around data which we have ready for future publication, but were not able to publish at the present time, or around tests now being made in connection with this year's program. After all, there was just about one half a heating season in which we were able to obtain the data included in the present report.

The principle questions raised were on the subjects of the radiator output, the radiator water temperatures (or, in fact, the water temperatures in the whole heating system), and the amount of water being circulated for the various test conditions. We do not have exact data on the radiator output, but we are making a laboratory

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setup in which the radiators can be calibrated to see if the output for a given average radiator water temperature is in accordance with those generally published in catalogs. The catalog ratings are the ones we used in selecting our radiation.

As for the water temperatures, about all that I can say now is that at no time did we experience water temperatures as high as 200 F. In other words, we were always under what was supposedly our design operating conditions. Of course, at no time did we experience an 80-deg indoor-outdoor temperature difference. This year we have had a few days in which the average outdoor temperature was about zero, which is close to our design conditions; and again we did not obtain an average radiator temperature of 200 F, nor did the circulator operate anywhere near 24 hours per day.

This seems to indicate further that the radiators are oversize. We have not yet determined the actual quantity of heat that is being put into the house from uninsulated basement piping and from our inside chimney, but we are working on these this year and will be able to report them in some future paper. The extrapolated circulator performance curves indicate that the radiators would be used at about one half their rated capacity when the outdoor air temperature is −10 F. The actual heat loss from the house under these conditions would be the sum of the heat put into the house by the radiators, the uninsulated heating pipes, chimney, occupancy, etc.

A question was raised regarding the maximum temperature variation between rooms. The temperature difference reported as 2.8 F is an average of a 24-hour period. All of the temperatures reported in this paper are for 24-hour test periods. The above value remained almost constant at all times in that we carried uniform house temperatures for the 24-hour period. The thermostat was not set back at night in any of these tests. The maximum temperature variation which has been determined in any room during a complete cycle of operation of the burner and circulator was in the neighborhood of one half to one degree.

The question was raised as to what proportion of the total power consumption was chargeable to the circulator. It so happens that the circulator power and the burner power consumption are split almost equally. Hence, the cost of operating the circulator represents about one half the total power consumption or about 5 per cent of the total cost of operation.

No. 1200

FACTORS INFLUENCING THE HEAT OUTPUT OF RADIATORS

By A. C. Davis,* W. M. Sawdon ** and David Dropkin †

Introduction

THE present ASHVE radiator test code (1927) in its description of the test room specifies that: "The test room shall be 12 x 15 ft in floor area and have a ceiling height of 9 ft with an allowable tolerance of plus or minus 10 per cent in any dimension. The walls and ceiling shall have a smooth, close surface and be painted flat gray in color. A window not larger than 16 sq ft will be permitted in the test room, but the window shall be shielded from direct rays of the radiator." [1]

It is evident from this description that a certain amount of freedom is permitted in the construction of the test room, the use of windows and the size of radiator. The main objective of the tests hereinafter described was to determine the effect of test room insulation, the number and arrangement of windows, and the size of radiator on the performance of household radiators.

Description of Laboratory and Apparatus

The Constant Temperature Room

The test room, in which each radiator when under test was located, was erected within the constant temperature room of the newly constructed heat transfer laboratory shown in Figs. 1 and 2. This laboratory consists of a constant temperature room, a cooler room, and an instrument room. The constant temperature room is 26 ft x 29 ft with a ceiling height of 12 ft. The cooler room is 7 ft 7 in. x 14 ft with a 12 ft 6 in. ceiling. The instrument room is 10 ft x 34 ft with a 12 ft ceiling.

The constant temperature room and the cooler room have a common wall, 12 ft x 14 ft; while the instrument room has a side common to both the con-

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** Professor of Experimental Engineering, Cornell University. Member of ASHVE.
† Instructor of Experimental Engineering and Westinghouse Research Associate, Cornell University.

[1] The Condensation Method for the Testing of Direct Radiators. (ASHVE Transactions, Vol. 33, 1927.)

Presented at the 48th Annual Meeting of the American Society of Heating and Ventilating Engineers, Philadelphia, Pa., January, 1942.

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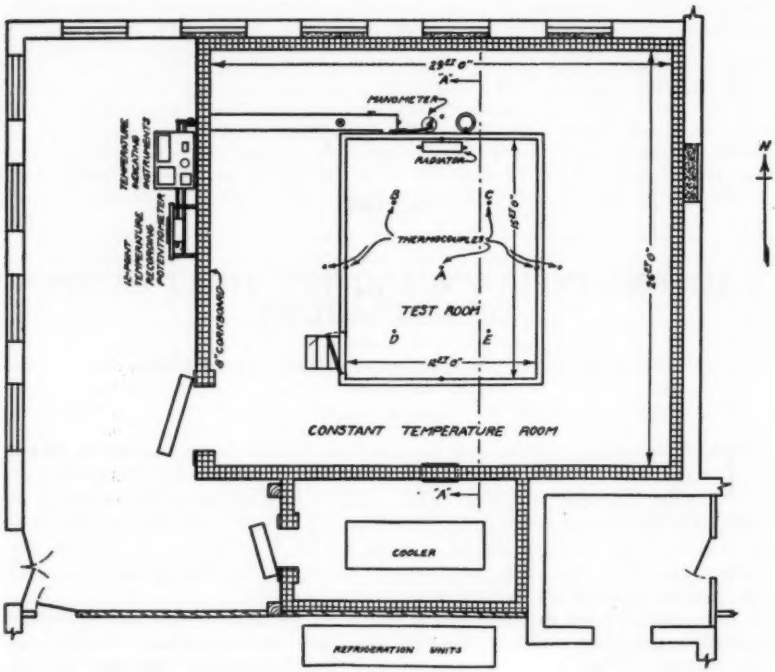

FIG. 1. PLAN OF HEAT TRANSFER LABORATORY

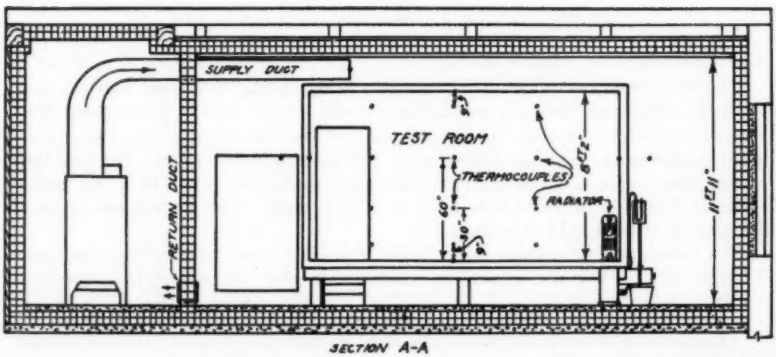

FIG. 2. ELEVATION OF HEAT TRANSFER LABORATORY

stant temperature room and the cooler room. The constant temperature room has a *super-freezer* door, 4 ft 6 in. x 6 ft and two *plug doors,* 2 ft x 2 ft each. Pipes, wires, etc., leading to the instrument room are introduced through the plug doors. The cooler room has a 2 ft 6 in. x 6 ft *super-freezer* door leading to the instrument room. All the walls of both the constant temperature room and the cooler room are insulated with 8 in. of cork applied in two layers. The floors have a base of 4 in. concrete upon which is placed 8 in. of cork and 3 in. of reinforced concrete. On the north side, the east end, and 11 ft of the south side of the constant temperature room the insulation rests against the brick walls of the building with 1 in. to 1½ in. of plaster between the walls and the cork. The windows are covered with masonite resting against the sash. The space between the masonite and the cork blocks is filled with ground cork. The ceiling of this room consists of a suspended steel deck, above which is 8 in. of cork with 1 in. of cement plaster on top of the cork. On the south side of the cooler room the insulation is placed against a 6 in. double-sided wooden partition. The inner surfaces of the walls of both the constant temperature room and the cooler room are plastered with asphalt emulsion and painted with a white cold storage paint. The walls, where exposed to the instrument room, are plastered on the outside with cement.

The cooling equipment has a capacity of 15 tons of refrigeration and consists of a 4 in. x 4 in. ammonia compressor, a 5¾ in. x 4 in. booster compressor, a 16 in. sixteen-pass condenser, a water cooled intercooler, a liquid and gas cooler, a liquid receiver, an oil separator, pressure gages, controls, etc. This equipment is located just outside the air cooler room. Within the cooler room is located the accumulator with float control and the cooler with its fan and motor.

The air which is supplied to the constant temperature room is cooled in the cooler room and then blown into the constant temperature room at the required temperature. This method of air cooling minimizes radiant heat losses from objects within the constant temperature room since the coldest object within the room is the air duct supplying the colder air. The duct is only slightly colder than the air of the room. This arrangement makes it possible to maintain a uniform temperature throughout the room by controlling the air distribution. During the radiator tests thus far conducted, the inlet air temperature has been about 6 F lower than the average air temperature within the constant temperature room.

The air temperature in this room is thermostatically controlled and the thermostat can be set to maintain any desired temperature between minus 50 F and plus 100 F. A carefully designed distributing system discharges the air near the ceiling in such a manner that at no time during a test was the difference in temperature, at the same level in different parts of the room, more than 1 F. The return opening is located near the floor.

EQUIPMENT AND METHOD OF TEST

The radiator test room was built to conform, in so far as it was possible, with the ASHVE test code specifications. The construction of the room was in sections so that it could be easily taken apart. The room as used for the tests hereinafter described is shown in Figs. 1 and 2. It was 15 ft long, 12 ft

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wide and 8 ft 2 in. high. Each section, except those used for the floor, consisted of a wooden frame made up of 1 in. x 4 in. studs placed on 16-in. centers. The frame was covered with ⅛ in. thick hard pressed masonite board. The floor was constructed in the same manner as the walls and ceiling except that 2 in. x 4 in. joists and ¼ in. thick masonite boards were used. A small section of the floor which supported the radiator was made of ⅞ in. thick wood flooring in order to provide extra strength and rigidity. The masonite wall back of the radiator was covered with ½ in. plaster board. The entire inside surface of the test room was painted with flat gray paint. To provide

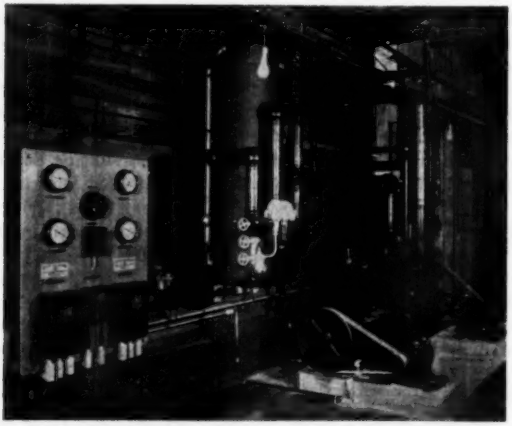

FIG. 3. AMMONIA COMPRESSORS USED TO PROVIDE COOLING FOR THE CONSTANT TEMPERATURE ROOM

easy access to the room a standard 1⅜ in. wooden door was located at the southwest corner of the room. This door, however, was never opened when an experiment was in progress.

Thirty copper-constantan thermocouples and five thermometers were installed to measure the air temperatures of the constant temperature room as well as the air and surface temperatures of the walls, floor, and ceiling of the test room. Air temperatures of the test room were measured at 9 in., 30 in., and 60 in. above the test room floor and at 9 in. below the ceiling, at five different stations (Figs. 1 and 2). The wall temperatures were measured 5 ft above floor level, while the floor and ceiling temperatures were measured at station *A*.

A 12-point recorder was used to record the air temperatures within the test room at the stations *A*, *B*, and *E*. The remaining thermocouples were connected to a semi-precision portable potentiometer and a complete set of readings was taken every 15 min.

In order to insure dry steam entering the radiator, the pipe line supplying the steam was heavily insulated; three continuous bleeders were provided, the last one being only about two feet away from the radiator. The steam enter-

TABLE 1*a*—RESULTS OF TESTS WHEN THE NUMBER OF SECTIONS IN A RADIATOR WAS VARIED

NUMBER OF SECTIONS IN RADIATOR	TOTAL E.D.R.	E.D.R. PER SECTION	INSIDE SURFACE TEMPERATURES OF TEST ROOM, DEG F						PER CENT DEVIATION IN RADIATOR OUTPUT PER SECTION
			North Wall	West Wall	South Wall	East Wall	Floor	Ceiling	
12	29.99	2.50	89.6	62.1	62.1	62.1	60.6	63.9	...
40	105.81	2.65	80.5	44.4	42.8	46.6	35.0	49.2	6.0

ing the radiator was always kept in a slightly superheated state. For all experiments reported in this paper the absolute pressure of the steam entering the radiator was kept at 31.9 in. of mercury. A well insulated condensate collector was attached at the outlet end of the radiator—thus completing a two-pipe system. During all test runs, the water level in the collector was kept high so as to minimize condensation. The pipe connecting the radiator to the collector was only about 6 in. in length and well insulated. The arrangement of all connections to the radiator was such that the condensation within piping was negligible as compared to the total condensation within the radiator. The condensate was drained continuously into pails. The surface of the water in the pails was covered with a layer of heavy oil to prevent a loss of condensate by evaporation. The condensate was weighed every 30 min on a calibrated scale which read directly to 0.01 of a pound. No test run was accepted unless the weight at each 30-min interval checked within less than 1 per cent. For all tests hereinafter described the desired temperature in the test room was maintained by controlling the temperature of the constant temperature room.

RESULTS OF TESTS

Effect of Size of Radiator

The present radiator test code does not specify the approximate size of radiator to be tested in the test room. It is obvious that the output per section of a

TABLE 1*b*—EFFECT OF INSULATION WHEN THE 5-FT LEVEL AT CENTER OF TEST ROOM WAS KEPT CONSTANT AT 70 F

NUMBER OF SECTIONS IN RADIATOR	TOTAL E.D.R.	E.D.R. PER SECTION	INSIDE SURFACE TEMPERATURES OF TEST ROOM, DEG F						PARTS OF TEST ROOM INSULATED
			North Wall	West Wall	South Wall	East Wall	Floor	Ceiling	
12	29.99	2.50	89.6	62.1	62.1	62.1	60.6	63.9	None
12	31.34	2.61	83.9	56.9	57.2	58.2	53.3	70.0	C
12	31.44	2.62	85.0	56.7	56.4	56.8	60.3	70.7	C, F
12	31.56	2.63	83.6	54.8	54.0	66.1	60.2	71.0	C, F, 1W
12	31.48	2.62	81.6	51.5	65.5	65.4	58.7	69.7	C, F, 2W
12	31.81	2.65	78.3	66.3	66.0	66.4	59.1	69.8	C, F, 3W

C = Ceiling. F = Floor. W = Wall.

TABLE 1*c*—EFFECT OF INSULATION WHEN THE 30-IN. LEVEL AT CENTER OF TEST ROOM WAS KEPT CONSTANT AT 66 F

NUMBER OF SECTIONS IN RADIATOR	TOTAL E.D.R.	E.D.R. PER SECTION	INSIDE SURFACE TEMPERATURES OF TEST ROOM, DEG F						PARTS OF TEST ROOM INSULATED
			North Wall	West Wall	South Wall	East Wall	Floor	Ceiling	
12	29.96	2.50	89.6	62.1	62.1	62.1	60.6	63.9	None
12	30.52	2.54	90.0	62.1	62.1	62.8	58.4	74.5	C
12	30.36	2.53	91.4	59.8	59.8	60.2	62.4	74.0	C, F
12	30.46	2.54	89.9	57.4	56.6	68.0	61.3	73.5	C, F, 1W
12	30.86	2.57	85.5	56.2	69.6	69.6	62.9	73.6	C, F, 2W
12	30.72	2.56	81.1	69.2	68.9	69.6	62.5	72.8	C, F, 3W
12	30.46	2.54	90.4	66.6	65.9	66.2	58.0	70.7	C, F, 4W

C = Ceiling. F = Floor. W = Wall.

large radiator will be higher than that of a small one when the same temperature at a given level is maintained in the test room. In order to determine the effects of size on radiator performance, the following experiment was made.

A 40-section, 5-tube, 26-in. small tube radiator, with sections on 1½-in. centers, was installed in the test room and its output per section determined. The size of the radiator was then reduced to 12 sections and the output per section was again determined. During both tests the temperature at the center of the test room at the 5-ft level was kept constant at 70 F.

Table 1*a* and Fig. 4 show the results of this experiment. From this table it may be seen that the larger radiator gave off 6 per cent more heat per section than the smaller one. This can be attributed to the fact that the larger radiator lost more heat per section by both convection and radiation.

Fig. 4 shows the temperature gradients at the center of the test room for the two sizes of radiator. These curves show that the air temperature below the 5-ft level was lower for the larger radiator. This lower air temperature resulted in greater heat loss per section by convection.

The heat loss per section by direct radiation was greater for the larger radiator as shown by the results given in Table 1*a*. Here it is shown that the

TABLE 1*d*—EFFECT OF WINDOWS IN THE TEST ROOM

NUMBER OF SECTIONS IN RADIATOR	TOTAL E.D.R.	E.D.R. PER SECTION	INSIDE SURFACE TEMPERATURES OF TEST ROOM, DEG F						REMARKS
			North Wall	West Wall	South Wall	East Wall	Floor	Ceiling	
12	30.46	2.54	90.4	66.6	65.9	66.2	58.0	70.7	No windows
12	30.64	2.55	91.1	66.7	66.1	66.4	58.7	71.3	Two windows, one at each end of radiator
12	31.15	2.60	...	64.8	63.7	64.1	56.5	68.9	Two windows, one back of radiator

surface temperatures of walls, floor, and ceiling were much lower when the 40-section radiator was tested than when the 12-section radiator was tested. The law of heat transfer by radiation states that the net heat transfer is directly proportional to the difference in the fourth powers of the absolute temperatures of the hot body and the cold body. The hot body, that is the radiator, was at the same temperature for both the 40- and the 12-section radiator tests while the cold body temperatures were different. It is obvious then in the light of these facts that the larger radiator lost more heat per section by radiation.

During the computation of the results of radiator tests, a question was raised as to whether the temperature at the center of the test room represented

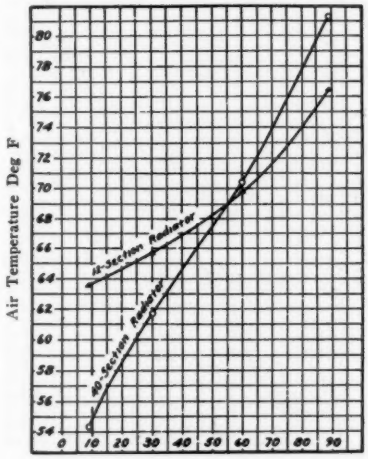

FIG. 4. COMPARISON OF TEMPERATURE GRADIENTS AT CENTER OF TEST ROOM FOR 40- AND 12-SECTION RADIATORS

the average temperature at a given level. The plot, in Fig. 5, clearly shows that for the test room used, the temperatures at the center of the room were approximately an average of the temperatures at a given level.

Effect of Test Room Insulation

In order to find the effect of insulation on both radiator output and temperature gradient within the test room, two series of tests were undertaken. For both series a 12-section, 5-tube, 26-in. radiator with sections on 1½ in. centers was used.

In the first series the temperature at the 5-ft level, in the center of the test room, was kept constant at 70 F. In the first test of this series the test room had no extra insulation, it being of the same construction as when the tests to determine the effects of size of radiator were made. In the second

test the ceiling was insulated; in the third test, ceiling and floor were insulated; in the fourth test, ceiling, floor and the east wall were insulated; in the fifth test, ceiling, floor, east, and south walls were insulated; and finally ceiling, floor, east, south, and west walls were insulated. In all of the above-mentioned tests the extra insulation used was 3 in. of mineral wool and ¼-in. plywood board.

The second series of tests was performed with the same variation of test room insulation as the first series. In this second series of tests a temperature of 66 F was maintained at the 30-in. level. An additional test was made with

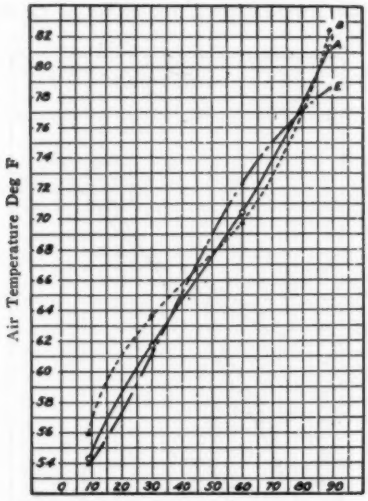

FIG. 5. TEMPERATURE GRADIENTS AT STATIONS A, B, AND E FOR THE 40-SECTION RADIATOR

the ceiling, floor, east, south, west, and north walls insulated. In this last test the north wall, that is the wall back of the radiator, was insulated so that its calculated value of over-all heat transfer coefficient was 0.24. This value corresponds closely to values commonly assumed for uninsulated homes.

The temperature gradients for the first series of tests, that is when the temperature at the center of the test room at the 5-ft level was kept constant at 70 F, are shown in Fig. 6. It may be seen from these curves that, when no extra insulation was used, the difference in temperature between floor and ceiling was less than under the other conditions of insulation. In general, when insulation was used, the temperature gradient remained approximately the same regardless of whether only the ceiling was insulated or the ceiling, floor, and the three walls were insulated. The surface temperatures, however, varied with the amount of insulation used. (See Table 1*b*.)

The heat loss through any section of the test room structure by conduction is equal to the heat transfer between the air and the inside surface of this structure. The surface transfer may be found by the following equation:

$$Q = A f_1 (t - t_1)$$

in which

Q = rate of heat transfer, Btu per hour
A = area of surface, square feet
f_1 = film coefficient, Btu per hour per square foot per degree Fahrenheit
t and t_1 = the temperatures of the inside air and the inside wall surface, respectively, degrees Fahrenheit

If a certain portion of the room is insulated, the inside surface temperature will increase. This will, in turn, reduce the amount of heat transferred from the air to this surface. For a given amount of heat to be transferred, it is obvious that the surface temperatures of the uninsulated portions of the room will have to be reduced in order to compensate for the smaller transfer of heat through the insulated sections.

As previously stated, heat loss from a radiator by convection depends on the temperature of the air which comes in contact with the radiator, if constant air velocity be assumed. When the ceiling of the test room was insulated, the temperature gradient became larger. When the temperature at the 5-ft level was kept constant, the temperatures below this level became lower when insulation was used. This means, then, that colder air came in contact with the radiator and thus a larger heat loss by convection occurred. However, as shown in Fig. 7, when the air temperature was kept constant at the 30-in. level, a better control of the temperature of the air coming in contact with the radiator was obtained. In general, the larger portion of the total heat loss from a radiator is by convection. For this reason one would expect a smaller effect of room insulation on radiator output when the temperature is controlled at the 30-in. level as compared with that at the 5-ft level. The foregoing statement is substantiated by data shown in Figs. 6, 7, and 8. Fig. 8 shows a direct comparison, in units of equivalent direction radiation,[2] of the radiator output per section, when the temperature was kept constant at the 60-in. level, with the output per section when the temperature was kept constant at the 30-in. level.

It should be observed that when the ceiling of the test room was insulated, the E.D.R. per section increased for both thermometer positions. In the case in which the 30-in. level was kept at constant temperature the increase was 1.6 per cent, while with the 5-ft level kept at constant temperature the increase was 4.4 per cent. The total increase of radiator output with the ceiling, the floor, and three walls insulated was 2.4 per cent for the 30-in. level and 6 per cent for the 60-in. level.

[2] The unit of E.D.R. is 240 Btu per hour.

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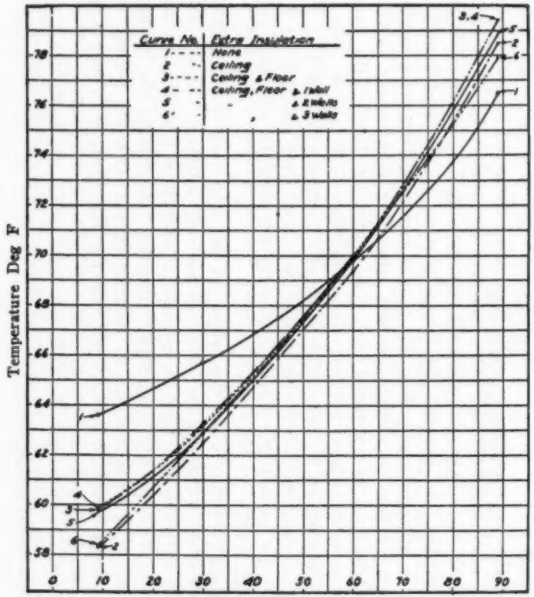
Curve No. | Extra Insulation
1 | None
2 | Ceiling
3 | Ceiling & Floor
4 | Ceiling, Floor & 1 Wall
5 | " " & 2 Walls
6 | " " & 3 Walls
Temperature Deg F
Height from floor in inches

FIG. 6. TEMPERATURE GRADIENTS AT CENTER OF TEST ROOM WITH DIFFERENT INSULATION CONDITIONS. TEMPERATURES KEPT CONSTANT AT THE 60 IN. LEVEL

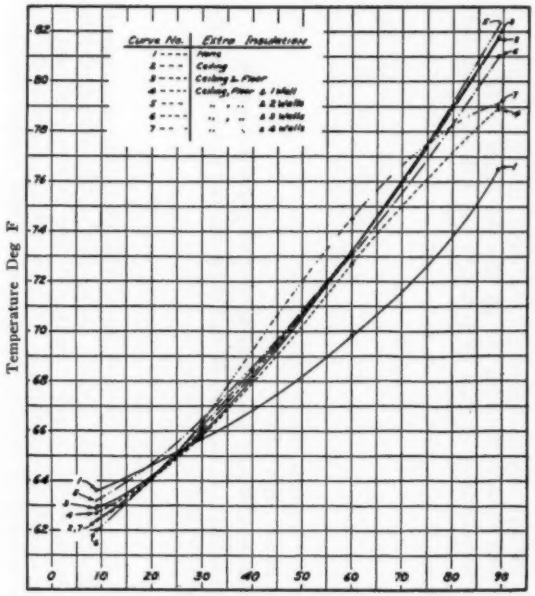
Curve No. | Extra Insulation
1 | None
2 | Ceiling
3 | Ceiling & Floor
4 | Ceiling, Floor & 1 Wall
5 | " " & 2 Walls
6 | " " & 3 Walls
7 | " " & 4 Walls
Temperature Deg F
Height from floor in inches

FIG. 7. TEMPERATURE GRADIENTS AT CENTER OF TEST ROOM WITH DIFFERENT INSULATION CONDITIONS. TEMPERATURES KEPT CONSTANT AT THE 30 IN. LEVEL

The curve in Fig. 8 shows that when the last wall was insulated (*i.e.* the one back of the radiator), the E.D.R. per section decreased. If the same amount of extra insulation had been placed on the wall back of the radiator as was placed on the other surfaces, the E.D.R. would have been the same as if no extra insulation had been used.

Additional tests were made using a 12-section, 6-tube, 25-in. radiator with 1¾ in. spacing of sections. This radiator was tested under the following test room conditions: First, with no extra insulation; second, with the ceiling insulated; and third, with the ceiling, floor, and three walls insulated.

The effect of adding the ceiling insulation was to increase the output of the radiator 4.8 per cent. When the ceiling, floor, and the three walls were insu-

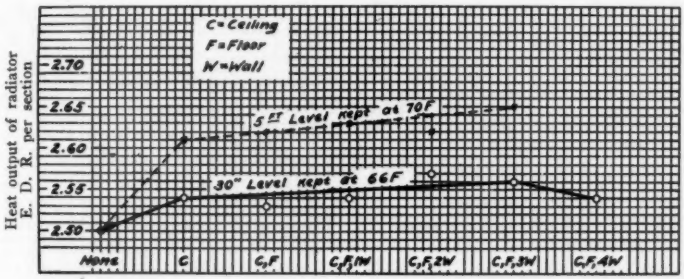

FIG. 8. COMPARISON OF HEAT OUTPUT OF A RADIATOR WITH DIFFERENT TEST ROOM INSULATION CONDITIONS, WHEN TEMPERATURE WAS KEPT CONSTANT AT 30-IN. LEVEL, WITH THAT WHEN TEMPERATURE WAS KEPT CONSTANT AT 5-FT. LEVEL

lated, the output was increased 8.6 per cent over that which was shown when no insulation was used.

Effect of Windows

The tests thus far described were made in a test room which had no windows. The ASHVE radiator test code discourages the use of windows although it permits a window not larger than 16 sq ft. An investigation of several test rooms in use at the present time reveals that many of them have windows.

In order to find the effect of windows on radiator heat output as well as on temperature gradient within the test room, the following experiments were performed:

1. A test was made when the test room contained no windows.
2. A second test was made with two windows, each 32 in. by 60 in. The radiator was installed midway between the windows.
3. A third test was made with one of the windows placed directly back of the radiator.

The windows were set in the north wall of the test room, the sill being 20 in. above the floor level. All cracks around the window sash were care-

fully sealed with tape to minimize leakage of air. The walls, floor, and ceiling of the test room were insulated with 3 in. of glass wool pads and ¼ in. plywood board. The wall back of the radiator was insulated in such a manner that its over-all heat transfer coefficient was 0.24.

The tests were conducted in the manner prescribed by the code except that the temperature at the center of the test room was kept constant at the 30-in. level instead of at the 60-in. level.

The results of these three tests are shown in Table 1*d* and in Fig. 9. It is evident that the effect of windows on radiator output and temperature gradient was negligible when the radiator was installed between the windows.

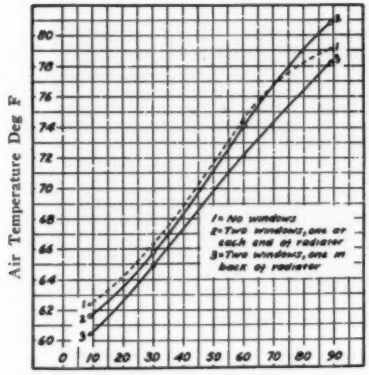

FIG. 9. COMPARISON OF THE TEMPERATURE GRADIENTS AT CENTER OF TEST ROOM WITH AND WITHOUT WINDOWS

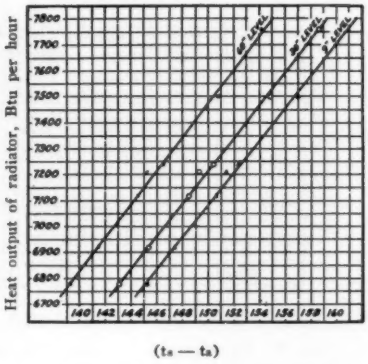

FIG. 10. RELATIONSHIP BETWEEN HEAT OUTPUT OF RADIATOR AND TEMPERATURE DIFFERENCE OF STEAM AND AIR AT DIFFERENT LEVELS

When one of the windows was placed directly back of the radiator, the difference in radiator output became more pronounced. For the radiator tested the output was increased 2.4 per cent above that which was shown when no windows were used. In this particular case the higher output was partly due to radiant heat loss to the cold window.

DETERMINATION OF EXPONENT *n* FOR THE CORRECTION FACTOR

In general, when testing radiators, it is difficult to maintain standard temperature conditions of steam and of air at the center of the test room. There are, however, two methods by which the radiator output can be brought to the standard conditions. One method is to obtain data on radiator output both above and below standard temperature conditions and then plot the data, as in Fig. 10. From the curve thus obtained, the output at standard conditions may be found. The second method is by using a correction factor.

The first method requires a great deal of time and several test runs to

obtain sufficient points for a plot. The second method requires but one run and is, therefore, preferable for ordinary testing. The correction is usually made as follows:

$$Q = Q_a\left(\frac{t_s - t}{t_s - t_a}\right)^n$$

in which

Q = heat given off by radiator at standard conditions, Btu per hour
Q_a = heat given off by radiator at the conditions of test, Btu per hour
t_s = temperature of the steam at standard conditions, degree Fahrenheit
t = temperature of the air at standard conditions, degree Fahrenheit
t_s = average temperature of the steam, degree Fahrenheit
t_a = average temperature of the air at center of test room at the required level, degree Fahrenheit
n = an exponent

$$\left(\frac{t_s - t}{t_s - t_a}\right)^n = \text{correction factor.}$$

The ASHVE radiator test code specifies that the steam pressure shall be one pound, the air temperature 70 F at the center of the test room at a height of 5 ft from the floor, and n, the exponent for the correction factor, 1.3. For these conditions, assuming that the barometric pressure is 29.92 in. Hg, which was no doubt intended, the correction factor becomes

$$\left(\frac{215 - 70}{t_s - t_a}\right)^{1.3} = \left(\frac{145}{t_s - t_a}\right)^{1.3}$$

From experience with tests made in the Mechanical Laboratories of the College of Engineering, Cornell University, in both commercial testing and research work it appears that the 30-in. elevation above the floor is preferable to the 60-in. for the control point of the test room temperature. This is clearly illustrated in Fig. 8 which shows only a small variation in radiator output when the 30-in. elevation was kept constant at 66 F, as compared with a larger variation in output when the 60-inch elevation was maintained at 70 F.

The 66 F temperature for the 30-in. elevation was selected because for room conditions showing the minimum temperature gradient a temperature of 66 F existed at the 30-in. level when the temperature at the 60-in. level was 70 F.

Tests were made to determine the correction factor which should be used when the temperature of the test room is maintained at the 30-in. elevation. In connection with these tests, information was obtained to verify the value of the exponent 1.3 for the 5-ft level. In the conduct of these tests the absolute pressure of the steam was maintained at 31.9 in. of mercury and data taken with the test room temperature constant at the desired elevation.

The results were plotted on both rectangular cross-section paper (shown in Fig. 10) and on log-log cross-section paper (shown in Figs. 11 and 12)

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with the radiator output in Btu per hour as ordinates and the coefficient C as abscissae.

Fig. 11 shows the results obtained by varying the room temperature between 61 F and 76 F at the 60-in. elevation. The slope of the line gives a value of $n = 1.33$. The graph in Fig. 12 shows the results of varying the room temperature between 56 F and 72 F at the 30-in. elevation. The slope of the line gives a value of $n = 1.31$. These results indicate that the value 1.3 is satisfactory for practical purposes within the limits given above.

A COMPARATIVE METHOD FOR RATING RADIATORS

Assuming that it is desirable to adhere to the condensation method, perhaps with some modifications, for rating radiators, it appears desirable to have an

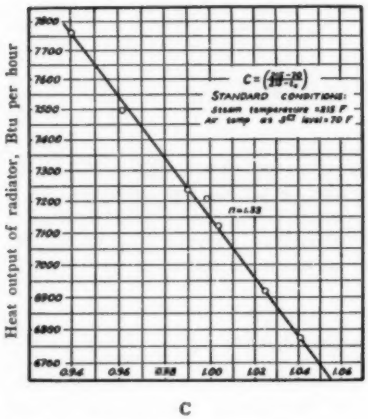

FIG. 11. LOG PLOT OF THE RELATIONSHIP BETWEEN HEAT OUTPUT OF RADIATOR AND COEFFICIENT C FOR THE 5-FT LEVEL

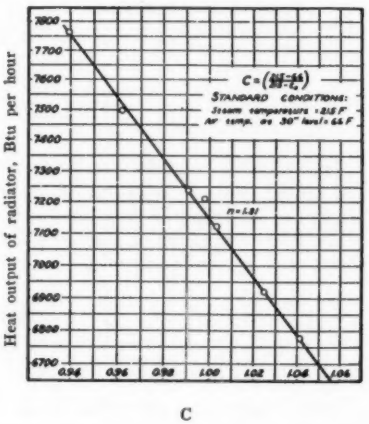

FIG. 12. LOG PLOT OF THE RELATIONSHIP BETWEEN HEAT OUTPUT OF RADIATOR AND COEFFICIENT C FOR THE 30-IN. LEVEL

alternative method for the use of manufacturers or even consumers who do not wish to install and maintain the equipment necessary to make tests as prescribed by the ASHVE code, but who have access to various sizes of radiators, which have been rated in the standard way, to use for comparison. This method can be used in the following manner: A radiator whose output is known is connected to the same steam line as the radiator whose output is to be determined. The ratio of their condensate weights is taken, then, as the ratio of their ratings.

The pipe connections to the two radiators should be symmetrical with respect to the steam line. These radiators can be tested in a large room regardless of its exact air temperature. The only precautions to be taken are that the

two radiators shall not be too greatly different in capacity nor in their general construction, that both radiators be similarly exposed to the cold surfaces within the room, and that the radiators be subjected only to natural convection of the air.

Tests made at Cornell with a parallel arrangement under three different conditions gave the following results: Under the first condition the radiators were placed in the center of a very large room and about 4 ft apart end to end. The ratio of radiator output of *A* to that of *B* was 1.011. The computed E.D.R. (equivalent direct radiation) for radiator *A* was 53.23 and for radiator *B*, 52.66.

These radiators were then set up in the same manner and tested in a completely enclosed 12 ft x 15 ft x 8 ft 6 in., galvanized iron room. The ratio of condensation of *A* to *B* was 0.985 while their respective E.D.R. were 70.29 and 71.41.

A third test was made using the galvanized iron room with the wall opposite the radiator removed. For this test the ratio of *A* to *B* was then 0.996 while their respective E.D.R. were 55.19 and 55.44.

The results show a maximum variation in the output of the two radiators of about 35 per cent, whereas the maximum variation in the ratio of the output of radiator *A* to that of radiator *B* was only 2.6 per cent.

SUMMARY OF RESULTS AND CONCLUSIONS

1. If two radiators of the same design are tested individually in the same test room, the one containing the larger number of sections will show the higher heat output per section. For tests made to demonstrate this difference the increase was 6 per cent.

2. Test room insulation affects the heat output of a given radiator. For the radiators tested, the largest increase in radiator output occurred when the entire room, except the wall immediately back of the radiator, was insulated. This increase was 6 per cent for the 12-section, 5-tube radiator and 8.6 per cent for the 12-section, 6-tube radiator.

Insulation of the ceiling accounts for the major portion of this increase. For the tests made under code specifications, ceiling insulation alone was responsible for 4.4 per cent of the total of 6 per cent increase which occurred when the floor, ceiling, and three walls were insulated.

3. The 30-in. level is a better height than the 60-in. level for the temperature control point of the test room. As shown in Fig. 8, with ceiling, floor, and three walls insulated the maximum increase in radiator output was 2.4 per cent when the temperature at the 30-in. level was kept constant as compared with a 6 per cent increase when the temperature at the 60-in. level was kept constant.

4. When infiltration of air was kept to a minimum, the installation of two windows in a standard test room had a negligible effect on radiator heat output.

The tests here reported show that there was practically no effect on radiator output when two windows were installed, with the radiator placed between them. However, when one of the windows was placed directly back of the radiator, the radiator heat output was increased 2.4 per cent.

5. The exponent, *n*, for the correction factor which is used to correct the results to standard conditions may be taken as 1.3 when the standard temperature is 66 F at the 30-in. level or 70 F at the 60-in. level.

6. A comparative method of test based on condensation can be used for the rating of radiators. This method is simple and with certain precautions should yield results well within 2.5 per cent accuracy.

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DISCUSSION

E. K. CAMPBELL, Kansas City, Mo.: There is one feature about this paper that interests me which pertains to the low temperature gradient within the room when the radiator is not operating at full capacity. I think it is true of any type heating equipment whether it be furnace or radiator, when it is operating at full capacity that it will heat the air that passes over it to a higher temperature, and thereby increases temperature gradient within the room.

The previous speaker questioned on the advisability of selecting and installing over sized radiators. Due to the fact that when a radiator frequently operates with a lower than design temperature, a reduction in room temperature gradient, in my opinion, should be taken into account when considering this question. In my judgment a larger radiator will function more economically if it has the proper controls installed to prevent over-heating.

W. M. SAWDON, Ithaca, N. Y.: It is true that the temperature gradient is one of the important factors. As was shown in this paper the greatest effect on temperature gradient was the insulation applied to the ceiling. It is impossible for me to state at this time how the effect of size of radiator might affect the temperature gradient of a room.

No. 1201

A COMPARATIVE STUDY OF FRICTION HEADS IN SCREWED AND WELDED ELBOWS

By F. E. Giesecke * and J. S. Hopper,** College Station, Texas

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the Texas A. and M. College.

DURING the study of friction heads in 6 in. pipes and elbows,[1] reported in a previous paper, it was decided:

1. To adopt the *no-length* elbow concept in calculating the friction heads in elbows.
2. To revise the friction head charts for pipes and elbows in accordance with the results of the 6-in. pipe and elbow test, the results of the earlier tests of smaller pipes and elbows, and the *no-length* elbow concept.
3. To continue the study of the relative friction heads in screwed and welded elbows.

No-Length Elbow

If, as shown in Figs. 2 and 6, the length of a pipe line is measured to and from the points of intersection of the center lines of the pipe line, a greater length is assigned to the pipe than actually exists and *no-length* remains to be assigned to the elbows.

If the results of friction head tests in pipes and elbows are recorded and reported on this basis, a smaller friction head is assigned to an elbow than would otherwise be the case, but the labor of calculating friction heads in pipe lines is reduced materially because the lengths of the pipe lines are measured to and from the points of intersection of their center lines, as shown in Figs. 2 and 6.

The revised friction charts for pipes and elbows are shown in Figs. 7 and 8 and need no detailed explanation. These charts are intended to be used in the design of pipe lines for hot water heating systems and not for the design

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** Assistant Professor of Mechanical Engineering, A. and M. College of Texas. Member of ASHVE.

[1] Friction Heads in Standard Six-Inch Pipe, by F. E. Giesecke and J. S. Hopper. (ASHVE Transactions, Vol. 47, 1941.)

Presented at the 48th Annual Meeting of the American Society of Heating and Ventilating Engineers, Philadelphia, Pa., January, 1942.

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of ordinary hydraulic installations. In hot water heating systems, properly operated, very little, if any, corrosion takes place because the same water is continually circulated through the system; for this reason, the charts of Figs.

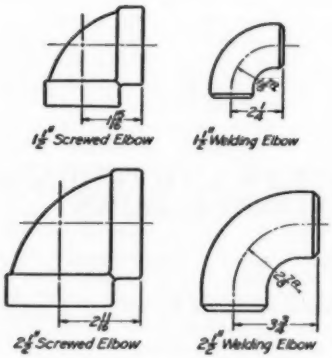

FIG. 1. FOUR TYPES OF ELBOWS TESTED

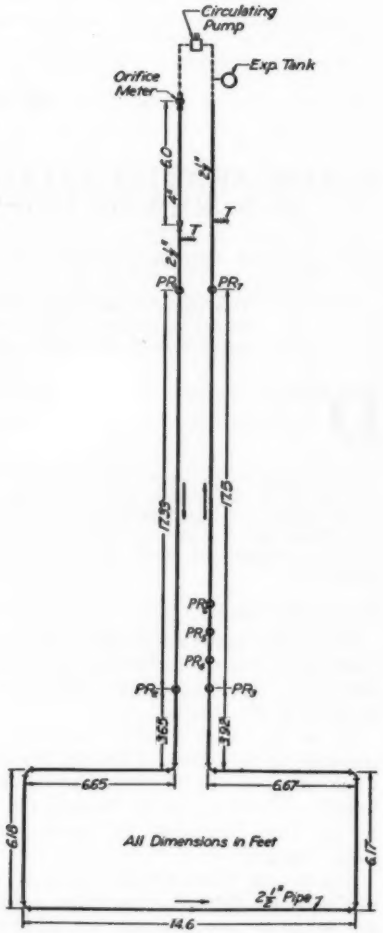

FIG. 2. (*right*) PIPE LINE FOR TESTING 2½-IN. ELBOWS

7 and 8 have only a small allowance for the increase in friction resulting from corrosion that may take place during the life of the system.

In ordinary hydraulic installations the amount of corrosion varies with the material of which the pipe is constructed, and with the quality and quantity

of water flowing through the pipe line. In such cases the amount of corrosion may be very material and larger friction factors must be used in the design of such installation than those shown in Figs. 7 and 8. During the study of the 6-in. pipe and elbow tests already referred to, it was found that the friction heads in welded 6-in. elbows were considerably lower than those in screwed 6-in. elbows; for example, at velocities of 4, 5, and 6 fps, the friction heads in the welded elbows were only 33, 37, and 40 per cent, respectively, or, on an average, 35 per cent of those in the screwed elbows.

Having found these ratios for 6-in. elbows, it was decided to determine the corresponding ratios for 2½-in. and 1½-in. elbows by laboratory tests, and

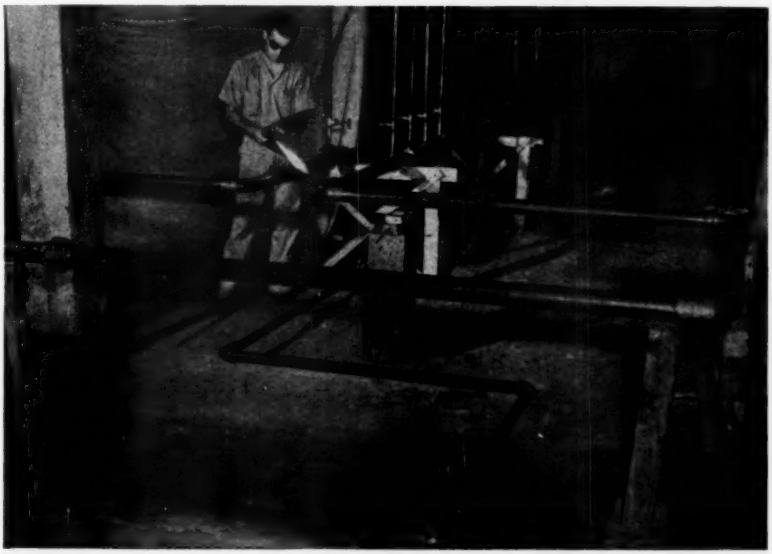

FIG. 3. PIPE LINE FOR TESTING 2½-IN. ELBOWS WHEN ABOUT TO BE REPLACED BY PIPE LINE FOR TESTING 1½-IN. ELBOWS

the corresponding ratios for other sizes by interpolation and extrapolation, if the ratios determined experimentally for 6-in., 2½-in. and 1½-in sizes should show a sufficiently definite relationship to permit that procedure.

The four types of elbows selected for this study are shown in Fig. 1. The experimental set-up for the 2½-in. elbow test is shown in Fig. 2. A portion of this set-up, when the 2½-in. welded elbow line was about to be removed and replaced by the 1½-in. screwed elbow line, is shown in Fig. 3.

The test procedure was the following. The water was circulated through the pipe line by a motor-driven pump; the flow of water in the pipe line was measured by means of a calibrated orifice-meter, located in a 10 ft section of 4-in. pipe; the friction heads in the several sections of the pipe line were

measured by the differences in the elevations of the water in vertical glass tubes, connected by pipe lines to the several piezometer rings shown in Fig. 2.

The pipe line section from PR_1 to PR_7 consisted of 82.67 ft of 2½-in. pipe and 6 screwed elbows; the section from PR_2 to PR_3 consisted of 47.84 ft of 2½-in. pipe and 6 screwed elbows. The difference between the two sections was 34.83 ft of 2½-in. pipe.

TABLE 1—TEST OF 2½-IN. SCREWED ELBOWS
(*July* 11 *and* 14)

METER READING INCHES	VELOCITY FEET PER SECOND	WATER TEMPERATURE FAHRENHEIT	FRICTION HEAD SECTION 1–7 INCHES	FRICTION HEAD SECTION 2–3 INCHES
1	2.60	84	15.0	11.0
1	2.60	85	17.0	11.8
1	2.60	90	16.0	10.5
1	2.60	91	14.5	10.5
Average		87.5	15.6	11.0
2	3.56	85	31.0	22.0
2	3.56	85	30.5	20.5
2	3.56	89	29.3	20.8
2	3.56	90	30.3	22.0
Average		87.3	30.3	21.3
3	4.41	86	44.0	31.5
3	4.41	86	45.0	32.0
3	4.41	89	43.3	31.0
3	4.41	90	43.5	31.0
Average		87.8	44.0	31.4
4	5.06	87	58.5	41.5
4	5.06	87	58.0	40.8
4	5.06	88	57.0	40.3
4	5.06	89	57.5	40.5
Average		87.8	57.8	40.8
5	5.70	87	72.3	50.8
5	5.70	88	71.5	50.5
Average		87.5	71.9	50.7

Section 1–7 consisted of 82.67 ft of 2½-in. pipe, 1 union, and 6 elbows.
Section 2–3 consisted of 47.84 ft of 2½-in. pipe, 1 union, and 6 elbows.

The results of the test of these two sections are shown in Table 1 and in Fig. 4, where the line AB ($h = 2.9V^{1.85}$) represents the friction heads in the section from PR_1 to PR_7 and the line CD ($h = 1.87V^{1.91}$) represents the friction heads in the section PR_2 to PR_3.

The differences between the friction heads shown by line AB and those shown by line CD are the friction heads in 34.83 ft of 2½-in. pipe.

Having thus determined the friction head in 34.83 ft of pipe, it was planned to use it to determine the friction head, first, in 1 ft of 2½-in. pipe, and second, in one 2½-in. elbow; before doing this, it was decided to check the results shown by lines AB and CD in Fig. 4 with those which should be expected according to the charts of Figs. 7 and 8, and it was found that for the higher velocities the measured friction heads were higher than those calculated from the charts and for the lower velocities they were lower.

According to the charts, the friction heads in one ft of 2½-in. pipe and in one elbow, and for velocities of 4, 5, and 6 fps are respectively, 0.330, 0.480, 0.660, 1.900, 2.800, and 4.000 in.

A second check was then made by calculating the friction heads in the two lines by the formulae [2] developed by the senior author for new and clean pipes, i.e.,

$$h = 0.00685 \frac{V^{1.79D-0.04}}{D^{1.275}} \text{ ft per foot of pipe}$$

$$\text{and } h = 0.0141 \frac{V^{1.96}}{D^{0.26}} \text{ ft per elbow.}$$

For 2¼-in. pipe these formulas become
$h = 0.0259_{7}{}^{1.73}$ in. per ft of pipe and
$h = 0.138_{7}{}^{1.96}$ in. per elbow.

The three sets of friction heads for the two pipe lines, from PR_1 to PR_7 and from PR_2 to PR_8, are shown in the following tabulations; the three sets for the line from PR_1 to PR_7 are also shown in Fig. 4.

VELOCITY—FEET PER SECOND

	4	5	6
Measured	37.7	57.90	80.00
Chart	38.6	56.48	75.56
Formula	36.1	54.08	75.32

Friction Heads for pipe line from PR_1 to PR_7.

Measured	26.4	40.4	57.3
Chart	27.18	39.76	55.57
Formula	26.17	39.48	55.29

Friction Heads for pipe line from PR_2 to PR_8.

It appears from these tabulations that the values derived from the chart agree well with those calculated according to the original formula; the values from the chart being slightly higher to allow for the effect of slight corrosion.

[2] University of Texas, *Bulletin* No. 1759.

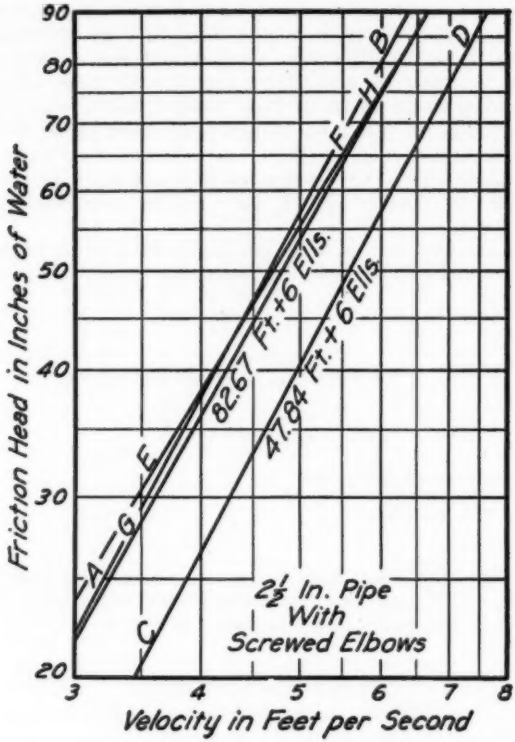

FIG. 4. FRICTION HEADS IN SECTIONS PR_3 TO PR_7 AND IN SECTIONS PR_2 TO PR_3 OF 2½-IN. PIPE LINE

The line *AB* was determined by experiment during this investigation. The line *EF* was calculated according to the values shown in the charts for Figs. 7 and 8. The line *GH* was calculated according to formulae determined in earlier studies.

Since the line *AB* is steeper than the lines *EF* and *GH*, there seems to be a disturbing element in the experimental set-up which increases the friction head in the 2½-in. pipe. This disturbing element may be the jet of water projected from the orifice meter into the pipe. When the velocity of the water in the 2½-in. pipe is 3 fps, the velocity of the jet is 5.98 fps; and when the velocity of the water in the 2½-in. pipe is 6 fps, the velocity of the jet is 11.95 fps. The influence of the jet must therefore increase with the velocity of the water and must tend to increase the slope of the line *AB* which represents the friction-heads in the 2½-in. pipe of the experimental set-up.

It also appears that the values secured by direct measurement do not agree with those derived from the chart. The cause for this disagreement is evidently that the jet of water which emerges from the orifice-meter into the 2½-in. pipe line with a high velocity creates additional turbulence which extends far into the pipe line and causes the additional friction head.

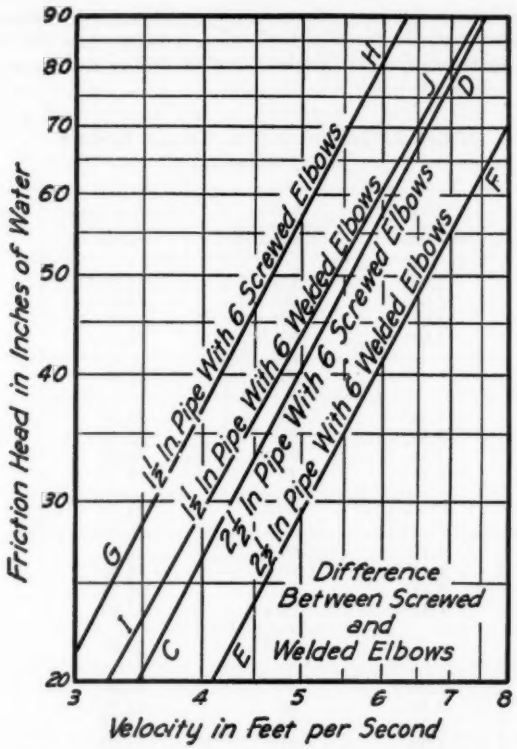

FIG. 5. DIFFERENCES BETWEEN THE FRICTION HEADS IN 6 SCREWED AND IN 6 WELDED ELBOWS FOR 1½-IN. AND 2½-IN. PIPE

The average slope of the two 2½-in. lines, *CD* and *EF*, is greater than was expected in comparison with the average slope of the two 1½-in. lines, *GH* and *IJ*. This may be the result of the influence of the jet of water which was projected from the orifice meter into the pipe lines, since, for any one velocity of the two pipe lines, the velocity of the jet is greater for the 2½-in. line than for the 1½-in. line. For example, for a velocity of 5 fps in the pipe lines, the velocity of the jet is 9.96 fps for the 2½-in. line and 4.23 fps for the 1½-in. line.

The two 2½-in lines diverge less than the two 1½-in. lines. This may also be the result of the influence of the jet of water from the orifice meter on the friction heads in the two pipe lines.

Studies are in progress to determine the influence of a jet of water on the friction head in a pipe line into which the jet is projected.

The orifice is 1.75-in. in diameter and hence the velocities of the jet of water are respectively 8, 10, and 12 fps when the velocities of the water in the 2½-in. pipe are 4, 5, and 6 fps or, the velocity of the water at the orifice is twice that in the pipe.

It has been known for some time that the additional turbulence produced within an elbow extends for a considerable distance into the down-stream pipe line and increases the friction head within that pipe line, but this is the

TABLE 2—TEST OF 2½-IN. WELDED ELBOWS
(*July 24 and 28*)

METER READING INCHES	VELOCITY FEET PER HOUR	WATER TEMPERATURE FAHRENHEIT	FRICTION HEAD SECTION 1–7 INCHES	FRICTION HEAD SECTION 2–3 INCHES
1	2.60	87	13.50	8.75
1	2.60	88	13.00	8.50
1	2.60	93	12.00	8.00
1	2.60	93	12.25	8.00
Average		90.3	12.69	8.31
2	3.56	88	23.75	15.00
2	3.56	89	25.25	16.50
2	3.56	93	24.00	15.00
2	3.56	93	23.75	15.50
Average		90.8	24.19	15.50
3	4.41	89	35.50	22.50
3	4.41	89	34.50	22.50
3	4.41	92	35.25	22.75
3	4.41	92	35.00	23.00
Average		90.5	35.06	22.69
4	5.06	89	46.00	29.50
4	5.06	90	46.00	30.00
4	5.06	91	45.75	29.50
4	5.06	91	45.50	29.00
Average		90.3	45.81	29.50
5	5.70	90	55.00	35.50
5	5.70	91	56.50	36.00
Average		90.5	55.75	35.75

Section 1–7 consisted of 82.58 ft of 2½-in. pipe, 1 union, and 6 elbows.
Section 2–3 consisted of 47.75 ft of 2½-in. pipe, 1 union, and 6 elbows.

first time the authors have experienced the disturbing effect of a stream of water projected into a pipe line.

It is evident from this study that the two lines, *AB* and *CD* of Fig. 4, should not be used to determine the friction head in one screwed elbow for comparison with that of one welded elbow, but that this friction head should be calculated by the formula $h = 0.138v^{1.98}$.

After the pipe line with the 6 screwed elbows, shown in Fig. 2, had been tested, the screwed elbows were removed and replaced by 6 welding elbows

in such a manner that the total length of the pipe line remained practically unchanged. The length of the section from PR_2 to PR_3 with screwed elbows was 47.84 ft and with welded elbows, 47.75 ft. The test was then performed with the welded elbows as it had been for the screwed elbows. The results for the section from PR_2 to PR_3 are shown in Table 2 and by line *EF* in Fig. 5. Line *CD* of Fig. 5 is the same as line *CD* of Fig. 4. The difference between the friction heads shown by line *CD* and those shown by line *EF*

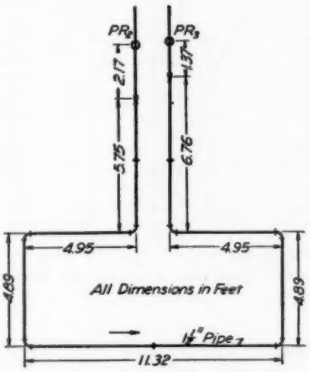

FIG. 6. PIPE LINE FOR TESTING 1½-IN. ELBOWS

are the differences between the friction heads in 6 *no-length* screwed and 6 *no-length* welded 2½-in. elbows.

The friction head in one *no-length* welded 2½-in. elbow was then calculated as illustrated in the following tabulation:

PIPE LINE SECTION	VELOCITY FEET PER SECOND		
	4	5	6
PR_2 to PR_3 ($h = 1.87\ V^{1.91}$) Line *CD*, Fig. 5	26.4	40.4	57.3
PR_2 to PR_3 ($h = 1.39\ V^{1.89}$) Line *EF*, Fig. 5	19.1	29.1	41.1
Difference between 6 screwed and 6 welded elbows	7.3	11.3	16.2
Difference between 1 screwed and 1 welded elbow	1.21	1.88	2.70
One screwed elbow ($0.138\ V^{1.96}$)	2.09	3.24	4.63
One welded elbow	0.88	1.36	1.93
Ratio	42%	42%	42%

To study the friction heads in 1½-in. elbows, the 1½-in. pipe line was attached to the 2½-in. pipe line used in the previous test, as shown in Fig. 6; the two pipe lines were joined by 2½ x 2 reducers and 2 x 1½ face bushings.

The pipe line section from PR_2 to PR_3 consisted of 3.54 ft of 2½-in. pipe,

TABLE 3—TEST OF 1½-IN. SCREWED ELBOWS
(*August 7 and 16*)

TIME SECONDS	WATER POUNDS	VELOCITY FEET PER SECOND	TEMPERATURE FAHRENHEIT	FRICTION HEAD SECTION 2–3 INCHES
300	500	1.89	88	9.00
60	100	1.91	86	9.25
59	100	1.93	86	9.25
290	500	1.96	88	9.37
39	100	2.97	86	20.75
38	100	2.97	86	20.50
183	500	3.10	88	22.75
178	500	3.19	88	24.00
30	100	3.79	86	33.00
30	100	3.82	86	33.50
134	500	4.25	88	41.50
26	100	4.35	86	43.00
26	100	4.35	86	42.90
130	500	4.37	88	44.00
109	500	5.19	88	60.00
22	100	5.20	86	58.00

Section 2–3 consisted of 3.54 ft of 2½-in. pipe, 2 reducers with face bushings, 43.51 ft of 1½-in. pipe, 3 unions, and 6 elbows.

2 reducers with face bushings, 43.51 ft of 1½-in. pipe, and 6 screwed 1½-in. elbows.

It was found that the orifice-meter, used in the previous test, could not be read with sufficient accuracy for the small flow needed for the 1½-in. line, so the pipe set-up was changed and the rate of flow through the 1½-in. pipe

TABLE 4—TEST OF 1½-IN. WELDED ELBOWS
(*August 22 and 27*)

TIME SECONDS	WATER POUNDS	VELOCITY FEET PER SECOND	TEMPERATURE FAHRENHEIT	FRICTION HEAD SECTION 2–3 INCHES
273	500	2.08	93	8.87
262	500	2.17	91	9.75
250	500	2.26	93	10.50
249	500	2.28	91	10.50
164	500	3.48	91	23.00
154	500	3.68	91	25.50
154	500	3.68	93	25.00
151	500	3.77	93	26.25
133	500	4.26	93	45.13
123	500	4.50	91	39.00
122	500	4.65	91	39.38
121	500	4.72	93	40.00
120	500	4.75	93	40.13
104	500	5.48	91	53.63
101	500	5.62	93	55.12

Section 2–3 consisted of 3.54 ft of 2½-in. pipe, 2 reducers with face bushings, 43.32 ft of 1½-in. pipe, unions, and 6 elbows.

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line determined by weighing the water; otherwise the test was performed in the same manner as that for the 2½-in. pipe. The length of the 1½-in. pipe line with screwed elbows was 43.51 ft with 6 elbows.

The results of this test are shown in Table 3 and by line *GH* of Fig. 5.

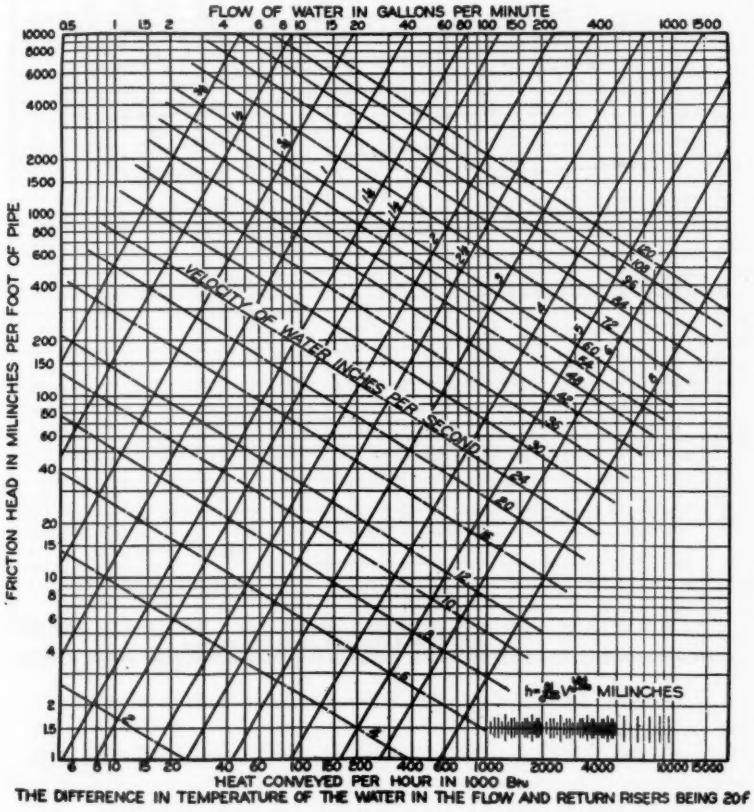
FLOW OF WATER IN GALLONS PER MINUTE
FRICTION HEAD IN MILINCHES PER FOOT OF PIPE
VELOCITY OF WATER INCHES PER SECOND
MILINCHES
HEAT CONVEYED PER HOUR IN 1000 Btu
THE DIFFERENCE IN TEMPERATURE OF THE WATER IN THE FLOW AND RETURN RISERS BEING 20F

FIG. 7. FRICTION HEADS IN STANDARD BLACK PIPE

The screwed elbows were then removed and replaced by welded elbows in such a manner that the length of the pipe line remained practically unchanged. The test was then performed as before; the results are shown in Table 4 and by line *IJ* of Fig. 5.

The friction heads in one *no-length* welded 1½-in. elbow were then calculated as illustrated in the tabulation as shown on page 212.

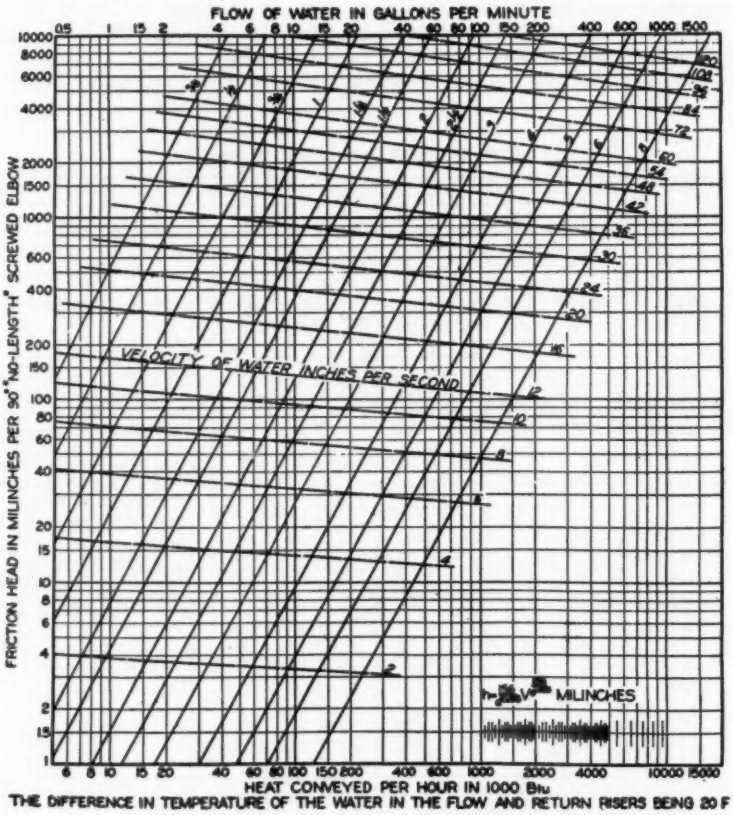
FLOW OF WATER IN GALLONS PER MINUTE
FRICTION HEAD IN MILINCHES PER 90° "NO-LENGTH" SCREWED ELBOW
VELOCITY OF WATER INCHES PER SECOND
MILINCHES
HEAT CONVEYED PER HOUR IN 1000 Btu
THE DIFFERENCE IN TEMPERATURE OF THE WATER IN THE FLOW AND RETURN RISERS BEING 20 F

FIG. 8. FRICTION HEADS IN NO-LENGTH BLACK ELBOWS (STANDARD)

CALCULATION OF FRICTION HEAD IN ONE NO-LENGTH WELDED 1½-IN. ELBOW

PIPE LINE SECTION	VELOCITY FEET PER SECOND		
	4	5	6
PR_2 to PR_3 ($h = 2.57\ V^{1.93}$)	37.3	57.4	81.6
PR_2 to PR_3 ($h = 2.46\ V^{1.79}$)	29.4	43.9	60.8
Difference between 6 screwed and 6 welded elbows	7.9	13.5	20.8
Difference between 1 screwed and 1 welded elbow	1.31	2.25	3.47
One screwed elbow ($0.149\ V^{1.93}$)	2.26	3.50	5.01
One welded elbow	0.95	1.25	1.54
Ratio	42%	36%	31%

SUMMARY

The ratios of the friction heads in *no-length* welded elbows to those in *no-length* screwed elbows, for the three velocities listed and for the three pipe sizes tested, are shown in the following tabulation.

PIPE SIZE	VELOCITY IN FEET PER SECOND		
	4	5	6
6-in	0.33	0.37	0.40
2½-in	0.42	0.42	0.42
1½-in	0.42	0.36	0.31

These results are not as consistent as had been expected before the investigation was begun. The variations in the results may be attributable to the fact that the three sets of elbows were tested by three different methods. For the 6-in. size, the results are based entirely on the experimental determinations. For the other two sizes, the ratios are based on the friction heads in new and clean screwed elbows as determined in earlier investigations. In the 1½-in. tests, the velocity of the water was determined by weighing the water flowing through the line; in the other two tests, the velocities were determined by means of orifice-meters.

However, it is possible with the *no-length* elbow concept, that the ratio may be independent of the velocity for the 2½-in. size; that it increases as the velocity increases, with the larger sizes; and that it decreases as the velocity increases, with the smaller sizes, as indicated by the results of this investigation.

CONCLUSION

As the final result of this study it appears that the ratio of the friction heads in *no-length* welded and *no-length* screwed elbows varies only slightly with the velocity of the water and also only slightly with the size of the pipe.

Until additional experimental data shall have been secured, it may be well to base friction head calculations on the friction heads in screwed elbows and to reduce the results to one-half, or slightly less, for welded elbows of corresponding sizes.

ACKNOWLEDGMENT

The authors take pleasure in acknowledging their obligation for valuable assistance in this study to L. P. Thompson, M.S., Instructor in Mechanical Engineering; B. R. Holland, M.S., Instructor in Chemistry; J. G. McGuire, M.S., Professor of Drawing; and H. P. Rigsby, Instructor in Welding. Mr. Thompson made practically all calculations; Mr. Thompson and Mr. Holland performed the major portion of the laboratory work; Professor McGuire prepared a majority of the drawings; and Mr. Rigsby did the welding.

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DISCUSSION

E. L. WEBER, Seattle, Wash. (WRITTEN): The advantages of welded connections over screwed connections in reducing friction are shown to be much greater than generally anticipated. Add to this a great number of efficient welders now being trained and we can look forward to the reduction of the use of screwed fittings and thereby improve the design of our piping systems.

W. R. RHOTON, Cleveland, Ohio: I would like to inquire of Dr. Giesecke what the maximum allowable water velocities are to preclude undesirable noise. I believe THE GUIDE states that water velocities of 11 fps constitute the upper limit for a building other than a factory. I was wondering whether you have observed in your laboratory work any undesirable velocities around 8 fps. Also I would like to know whether the old theory still applies that the friction head increases with the square of the velocities.

QUESTION: In Fig. 7 of the paper the pipe sizes are shown from ⅜ to 8 in. Are these actual inside diameters or are they standard pipe sizes?

DR. GIESECKE: Answering Mr. Rhoton, we have had no experience with velocities high enough to produce noise. As to the rate at which friction heads increase with velocity, this depends on the pipe size and that in turn depends on the relative roughness of the internal surface of the pipe. For the 2½-in. pipe, we found the fact that it varied as the 1.81 power of the velocity. If the internal surface of a 6-in. pipe has the same absolute roughness as that of a 3-in. pipe, the relative surface roughness of the 3-in. pipe is greater than that of the 6-in. pipe, and the exponent of V in the equation $h = kV^n$, is smaller for the 6-in. pipe than for the 3-in. pipe. This exponent varies, approximately, from 1.9 for ⅜-in. pipe to 1.76 for 12-in., new and clean standard iron or steel pipe.

Regarding the friction head charts, the sizes marked thereon are the nominal sizes of standard black pipe, but the letter d, in the formula, represents the actual internal diameter of the pipe, in inches. The charts apply only to standard pipe; they do not apply to heavier pipe because the internal diameters of such pipe are smaller than those of standard pipe. The statement that the friction head in pipe varies as the square of the velocity is only approximately correct. For new and clean iron or steel pipe the friction head varies more nearly as the 1.83 power of the velocity, on an average.

No. 1202

DESIGN AND PERFORMANCE OF A DIRECT-FIRED UNIT HEATER

By R. M. Rush * and H. A. Pietsch,** Pittsburgh, Pa.

DIRECT-FIRED unit heater applications are comparatively new in American industry. With the advent of steam unit heaters, which were introduced about 1918 and came into general use about 1926, industrial heating problems were greatly simplified. As the unit method of heat distribution came into wider use it was natural that the demand came for an efficient direct-fired unit for places where steam was not available or was not economical. The requirements of such a unit were that it be suitable for suspension if necessary, or if of the floor type, that it occupy comparatively small floor areas; that it be capable of firing at good efficiency either with oil or with gas, and that it be dependable and safe to use. The purpose of this paper is to trace the development in the designs of some large direct-fired heaters.

Cast-iron and welded steel warm air furnaces were of course in general use, but because of their large space requirements and their low efficiency, were not suitable for industrial unit heater applications. In 1933 when the development to be described was started, unit heaters firing natural gas were in use, but in general they were limited to the suspended type and to small capacities (of the order of 100,000 to 300,000 Btu per hour), and were not suitable for firing with fuel oil. An unfilled need therefore existed for a compact unit which would be suitable for either oil or natural gas firing.

Thus in 1933 there was little experience on which to base a design for 20 oil-fired units, each having an output capacity of 750,000 Btu per hour, for use in the sheet storage building of a steel company. Fig. 1 shows the design first adopted. The combustion chamber was made of corrugated sheet steel in the shape of an inverted U. Fans located near the floor caused air to circulate upward, over the rounded top of the chamber and then downward, discharging at the bottom of the other side as shown in the illustration.

The combustion chamber was lined about half-way up, with insulating fire brick, and the gases of combustion were drawn down on the outside of this lining, through the space between it and the corrugated steel shell, to a central flue running lengthwise of the bottom of the heater, by means of an exhauster

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Presented at the 48th Annual Meeting of the American Society of Heating and Ventilating Engineers, Philadelphia, Pa., January, 1942.

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mounted on the same shaft as the blower fans and of course driven by the same motor.

Early in the experiments with this unit, which was oil fired, it was found that the heating surface, consisting of plain corrugated sheet steel, was very apt to overheat and to burn up. After many trials of various means for overcoming this trouble, the problem was solved by the means shown in Fig. 5. This consisted in welding to the corrugated sheet shell a multiplicity of vertical fins, each fin having a vertical deflector bent back toward the outer casing. This construction acts in a number of ways to increase the heat abstraction from the corrugated sheets. It might be thought that the only effect of the deflectors is to increase the velocity of the air, but such is not the case; their

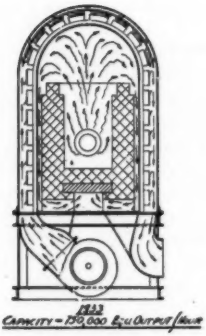

FIG. 1. CROSS-SECTION OF HEATER AS BUILT IN 1933

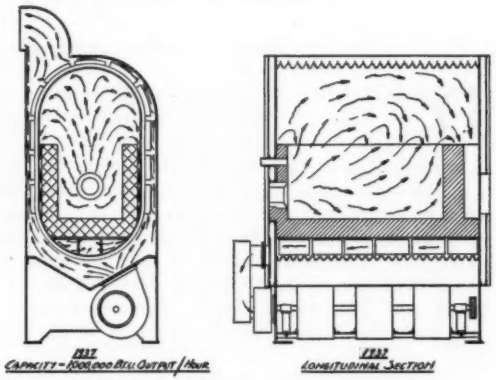

FIG. 2. CROSS-SECTION AND LONGITUDINAL SECTION OF HEATER AS BUILT IN 1937

chief function is to hold that air in contact, or throw it back into contact with the corrugated sheet and to force it to flow in the corrugations, especially when passing over the rounded top where the air tends to hug the outer casing. The deflectors also increase the turbulence of the air stream at the place where it is most effective for wiping off the stagnant film of air from the corrugated sheet; it is thought that vortex sheets are formed in the air stream at the trailing edge of each deflector, and are forced into contact with the corrugated surface by the air stream constrained by the succeeding deflector. The fins, being welded to form a metallic joint with the corrugated sheet, conduct heat outward and dissipate it to the air stream; calculations based on the theory of heat conduction show that the fin surface should be about 55 per cent to 60 per cent as effective per unit of area as the corrugated surface. The deflectors also act as secondary heating surfaces, receiving heat by radiation from the sheet and dissipating it to the flowing air; in doing so, they act as radiation shields with respect to the outer casing, which accounts for the very low percentage of heat loss from the latter.

That a high velocity of the air was not of itself sufficiently effective was

shown by a trial in which the same fins were used but the deflectors were omitted and the outer casing was moved closer to the corrugated sheet; although the true air velocity was as high or even higher than with the deflectors, overheating of the sheet persistently occurred at some places, notably in the curved top part.

One does not ordinarily think of heat transfer to a gas such as air as being at all of the same order of magnitude as heat transfer to a liquid such as water, as for example in boilers; hence it is of interest to note that with the fins and deflectors the average heat transfer attained the value of approximately 6000 Btu per square foot per hour, which is of the same order as that usually attained in low pressure steam heating boilers.

Heaters of the original design, Fig. 1, discharged the warm air at the floor level. This was considered advisable for the mill storage buildings as it would keep the steel sheets warm and free of condensation, but was not suitable for locations where the operators would be in the path of the warm air stream. Fig. 2 shows a design similar to the conventional floor type steam unit in that the cold air was taken from the floor and the warm air discharged through adjustable nozzles with louvers approximately 10 ft above the floor line. As compared to the original design, it will be noted that the bottom of the combustion chamber was made rounded the same as the top, and was air-swept; thus the effective heating surface was increased without increasing the over-all dimensions.

The combustion chamber was located off center in order to equalize the air resistance on the two sides.

Because of the difficulty of getting the air to scrub effectively that portion of the heating surface at the top where the air stream from the two sides converge and turn to enter the outlet nozzles, this part of the combustion chamber was very sensitive to overloads, and sometimes due to improper burner adjustment combined with overloads the corrugated sheet at this point was actually destroyed.

Other objections were with width of the unit, and the comparatively high power consumption. Due to the necessity for having insulating fire-brick walls in the combustion chamber at least 9 in. thick on each side, the width could not be reduced below 4 ft, plus 6 or 8 in. for the panel board, and in many places this width was quite objectionable. Due to the lack of streamlining of the combustion chamber, the friction loss was high, and the power required was 7½ hp as compared with 5 hp for a steam unit heater of the same capacity, namely, 1,000,000 Btu per hour.

In the fall of 1939, the design was further improved as shown in Fig. 3. Tube surface in the form of an economizer or air preheater section was added just above the outlet of the fans. Extensive tests were run on the effect of the tubes in cooling the gases and on the effect of twisted steel strips located in these tubes, known as *twisters*. These broke up the warm core of the gases, forced them to the outside of the tubes, and increased their turbulence in contact with the tube surface; increasing the heat transfer rate to such an extent that the exit temperatures were reduced by an additional 90 to 100 F. The latter is a more significant figure than at first appears, since 100 deg drop is relatively difficult to obtain at gas temperatures as low as 400 or 500 F for the reason that the *heat head* or temperature difference between the gases and the air is comparatively small.

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With this design, efficiencies exceeding 80 per cent were attained in three units built to heat a hangar for the U. S. Navy, at an output of 1,500,000 Btu per hour each, when fired with heavy oil.

In view of the favorable results obtained with the combination of tube surface with the conventional heating surface, a program of research was inaugurated to develop a heater combining tubes and a refractory lined combustion chamber that would overcome the objections to the previous model. In the spring of 1941, the design shown in Fig. 4 was evolved. The corrugated heating surface with fins and deflectors was of course retained, but the combustion chamber was streamlined in shape. The thick fire-brick lining was re-

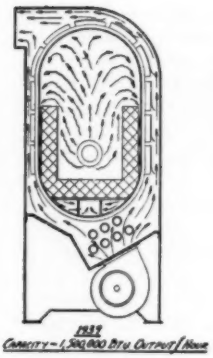

FIG. 3. CROSS-SECTION OF HEATER AS BUILT IN 1939

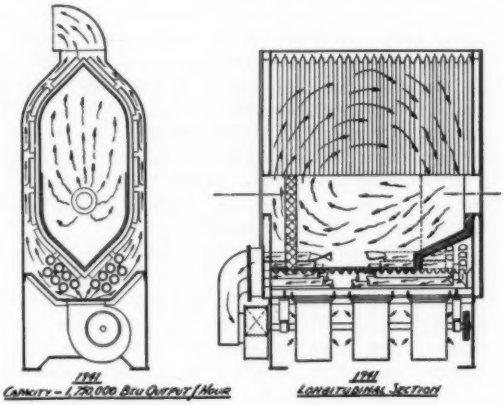

FIG. 4. CROSS-SECTION AND LONGITUDINAL SECTION OF RE-DESIGNED HEATER AS BUILT IN 1941

placed by a thin monolithic shell of plastic refractory, in intimate contact with the corrugated sheet, the refractory shell being thin enough to transfer a considerable amount of heat by conduction. Thereby the width of the combustion chamber and of the unit over-all could be reduced at least 12 in., and the narrower combustion chamber could be more easily streamlined to decrease the air resistance. Even with the tubes interposed in the path of the air, it was found in exhaustive tests that the over-all resistance to air flow was so lessened that a 5 hp motor could be used on a heater having an output of 1,000,000 Btu per hour, as compared to 7½ hp in the older design.

By the use of the pointed top in the combustion chamber, the dead space where the air stream from the two sides converged was eliminated, and all parts of the surface were scrubbed by turbulent air; at the same time the intensity of the radiation per unit of area at this place was decreased, both because the peak is farther from the flame and because of the inclination of the surface (Lambert's Law). Furthermore, impingement of the flame is less likely to occur in the *V* than on a broad rounded surface. As a result

of all these factors, over-heating at the point of air outlet was entirely overcome.

This design provided such other advantages as: avoidance of short-circuiting of the gases to the outlet; easier cleaning of convection surfaces; the possibility of installing the refractory lining at the factory rather than in the field; and reduction of heat storage in the refractory, with decreased danger of overheating the metal when the fans shut down. So much heat was stored in the 1700 lb of insulating firebrick of the older design that the fans had to be operated long after the fuel burner was shut off in order to prevent overheating of the corrugated sheet; and electric power was wasted thereby.

In the 1941 design, the smaller weight of refractory makes it possible to operate the blower fans, off and on with the fuel burner, without danger of

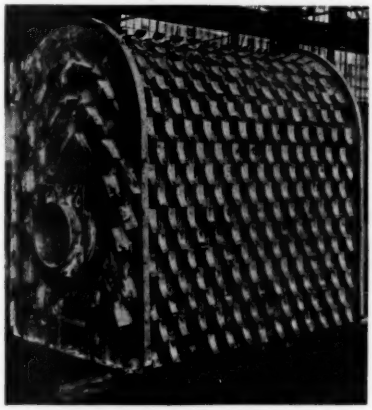

FIG. 5. CORRUGATED SHELL WITH FINS AND DEFLECTORS (1933 MODEL)

overheating the metal. Automatic spark ignition for the gas pilot prevents the latter from being extinguished if the combustion chamber becomes filled with products of combustion after the heater has shut down; and the change of location of the instrument and switch panel, as in Fig. 6, meant a reduction in heater width of 8 in., making the over-all width 3 ft.

In a heater of this type the refractory lining becomes glowing hot, which insures continuous ignition of the flame, and keeps the latter out of contact with cold surfaces, and by its radiation speeds up combustion. Since combustion takes place within a gas-tight welded steel chamber, under conditions of carefully controlled draft, there is little chance for infiltration of excess air. The conditions are equally good for the combustion of either gas or oil, and either fuel can be used by a simple change of burner, or a combination gas and oil burner may be used. For natural gas, standard equipment consists of a premix type of burner taking air from the blowing fans of the unit. In the case of oil burners, if light oil is to be used, a standard

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pressure type burner, with electric ignition, is used. For heavy oil, it was formerly necessary to use combination gas and electric ignition to insure automatic operation, but within the last two years development work in collaboration with a burner manufacturer has resulted in an oil burner in which only electric ignition is required. An electric resistance heater is embodied in the burner tube, and this keeps the oil hot up to the burner tip. The fuel is atomized by compressed air from a small rotary compressor integral with the burner. It is thus only necessary that heavy oil such as Bunker *C* be furnished at a temperature allowing it to flow freely; the burner then handles it automatically with simple electric ignition. By means of the compressed air, modulating pneumatic control is obtainable on the burner, which in the

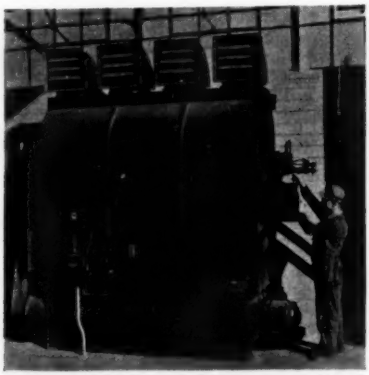

FIG. 6. 1,000,000 BTU UNIT WITH PANEL BOARD AT SIDE

case of heavy oil is more desirable than off-and-on control, the thermostat modulating the fuel supply to a rate just sufficient to hold the required temperature.

Under usual conditions of installation of heaters of this type, the radiation loss or the heat dissipated from the outer casing is not truly a loss, since it is utilized for heating the building. Hence, the only losses are the sensible and latent heat in the flue gases. Even when the conditions are such that the heat dissipated from the outer casing is not utilized, the loss is small, since previous tests of a similar heater showed that it did not exceed 0.8 per cent of the heat input.

Testing of the Re-Designed Heater

Early in 1941 a laboratory was set up in Pittsburgh and a 750,000 and a 1,000,000 Btu heater were installed. Arrangements were made for testing with both oil and gas. Daily test runs were made for several weeks during the summer. A full mechanical crew with electric welding equipment was available in order to make changes quickly. Observation openings were pro-

vided, by removing small sections of the outer casing and replacing with glass, giving the effect of windows. Twelve such observation ports on each heater were suitably located to permit observation of those points on the corrugated shell which previous experiences indicated were likely to become the hottest. The heaters were run continuously at overloads of as much as 30 per cent above their rated capacity, but observations indicated no hot spots in the surface surrounding the combustion chamber. Even distribution of air flow over the heating surface is essential in this type of heater, and the

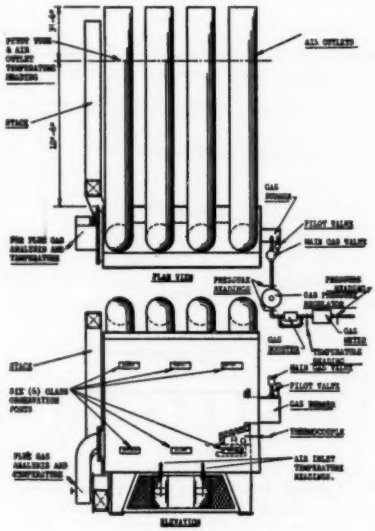

FIG. 7. DIAGRAMMATIC SKETCH OF TESTING ARRANGEMENTS, SHOWING LOCATION OF INSTRUMENTS

absence of overheated spots indicated that this objective had been attained in the re-designed heater.

The test set-up was as shown diagrammatically in Fig. 7. The quantity of air being heated was measured by Pitot tube traverses in four 16-in. diameter ducts leading horizontally from the four outlets of the heater. The distances from the Pitot tube location to the outlet of the heater and to the open end of the duct were somewhat in excess of the minimum prescribed in the standards of the *American Society of Mechanical Engineers.* The traverses were made according to the standard method, two across each duct at 45 deg to the horizontal and at 90 deg to each other; the 10-point method being used, with zones of equal area. Temperature readings at the heater outlets were erratic because of the turbulent conditions there, and for this reason the final air temperatures were taken at the same place as the Pitot tube traverse.

Ten readings of temperature were made in each outlet duct, at the same stations as those for the Pitot tube readings, in order to determine the average. Calibrated glass thermometers were used for these as well as for the inlet temperatures. The average of eight inlet air temperatures was taken as the true value.

The air quantity was also checked by two other methods, namely by use of the characteristic curves supplied by the fan manufacturer, showing cubic feet per minute for given shaft input horsepower and revolutions per minute; and by calculation from the heat balance and the measured air temperature

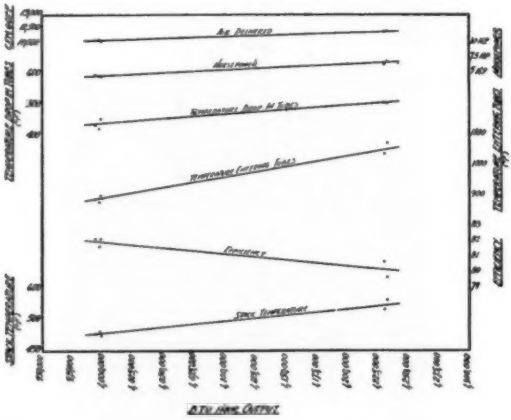

FIG. 8. CHARACTERISTIC CURVES 1941 UNIT HEATER SHOWING RESULTS OF TESTS

rise, involving determination of the heat losses from the casing and in the flue gases.

The quantity of natural gas burned was measured by a calibrated displacement meter. The temperature and pressure of the gas were measured at the meter, and the volume corrected to standard conditions. The analysis and the calorific value of the gas were supplied by the local gas company.

The temperature of the gases leaving the heater was measured by a mercury thermometer, and Orsat analyses were made of samples taken at this point, determining CO_2, O_2, and CO. The temperature of the gases entering the economizer was measured by a thermocouple and potentiometer.

The motor efficiency data were furnished by the manufacturer and the amperes input, voltage and power factor were determined by instruments furnished by that company. Motor speed and fan speed were taken with a revolution counter.

Pressures (or drafts) were measured in the combustion chamber, before the economizer, and in the outlet stack.

TABLE 1—HEAT BALANCE DATA FOR 1,000,000 BTU OUTPUT TEST

MEASURED VALUES

1	Heating value of gas, Btu per cubic foot at 30 in. Hg and 60 F	1128
2	Specific gravity	0.658
3	Fuel consumption as metered in cubic feet per hour	1142
4	Temperature of gas at meter	91 F
5	Pressure of gas at meter "inch water"	0.925
6	Barometer reading "inch Hg"	29.15
7	Exhaust gas analysis	Per Cent
	(*a*) CO_2	10.8
	(*b*) O_2	2.1
	(*c*) CO	0.0
8	Temperature of gases at exhauster	450 F
9	Temperature of air at inlet to fans	89.7 F
10	Temperature of air at discharge ducts approximately 11 feet beyond heater outlet	172.5 F
11	Voltage at fan motor	234
12	Amperes at fan motor	14.4
13	Power factor	0.75
14	Motor Efficiency (from manufacturers' data)	85%
15	Fan speed, revolutions per minute	928
16	Average reading at Pitot tube in outlet ducts (inches of water)	0.2695

CALCULATED VALUES

1	Volume of gas burned, corrected for meter calibration, cubic feet per hour referred to 60 F and 30 in. Hg	1052
2	Heat input in fuel, Btu per hour	1,186,000
3	Pounds dry gas, per pound of fuel at exhauster	17.15
4	Horsepower used by fans	4.98
5	Air delivered, as measured by Pitot tube, cubic feet per minute referred to 60 F and 30 in. Hg	10,625
6	Air delivered, according to characteristic curve of fan manufacturer, cubic feet per hour referred to 60 F and 30 in. Hg	10,890
7	Temperature rise of air through heater	82.8 F
8	Heat imparted to the air, Btu per hour.	
	(*a*) Based on Pitot tube air volume	968,000
	(*b*) Based on volume air from fan curve	993,000

HEAT BALANCE

	ITEM	BTU PER HOUR	PER CENT HEAT
1	Heat input in fuel	1,186,000	100
2	Heat loss due to water from combustion of hydrogen (latent plus sensible heat)	131,200	11.08
3	Heat loss due to dry chimney gases	78,700	6.64
4	Heat loss due to unburned combustible	0.0	0.0
5	Useful heat from fuel Items No. 1 minus 2 + 3 + 4	976,100	82.28
6	Heat imparted to the air based on volume as read by Pitot tube and air temperature rise	968,000	81.6

TABLE 2—HEAT BALANCE DATA FOR 1,250,000 BTU OUTPUT TEST

MEASURED VALUES

1	Heating value of gas, Btu per cubic foot at 30 in. Hg and 60 F.	1128
2	Specific gravity	0.658
3	Fuel consumption as metered in cubic feet per hour	1485
4	Temperature of gas at meter	90.75 F
5	Pressure of gas at meter (inch water)	3.2
6	Barometer reading (inch Hg)	29.06
7	Exhaust gas analysis	Per Cent
	(*a*) CO_2	11.2
	(*b*) O_2	1.8
	(*c*) CO	0.0
8	Temperature of gases at exhauster	533.5 F
9	Temperature of air at inlet to fans	87.1 F
10	Temperature of air at discharge ducts approximately 11 ft beyond heater outlet	181.3 F
11	Voltage at fan motor	225
12	Amperes at fan motor	17.9
13	Power factor	0.836
14	Motor efficiency (from manufacturers' data)	85 per cent
15	Fan speed, revolutions per minute	1033
16	Average reading at Pitot tube in outlet ducts (inches of water)	0.3474

CALCULATED VALUES

1	Volume of gas burned, corrected for meter calibration, cubic feet per hour referred to 60 F and 30 in. Hg	1350
2	Heat input in fuel, Btu per hour	1,523,000
3	Pounds dry gas, per pound of fuel at exhauster	16.55
4	Horsepower used by fans	6.64
5	Air delivered, as measured by Pitot tube, cubic feet per minute referred to 60 F and 30 in. Hg	11,900
6	Air delivered, according to characteristic curve of fan manufacturer, cubic feet per hour referred to 60 F and 30 in. Hg	11,780
7	Temperature rise of air through heater	94.2 F
8	Heat imparted to the air, Btu per hour	
	(*a*) Based on Pitot tube air volume	1,232,000
	(*b*) Based on volume air from fan curve	1,218,000

HEAT BALANCE

	ITEM	BTU PER HOUR	PER CENT HEAT
1	Heat input in fuel	1,523,000	100
2	Heat loss due to water from combustion of hydrogen (latent plus sensible heat)	174,200	11.43
3	Heat loss due to dry chimney gases	120,500	7.91
4	Heat loss due to unburned combustible	0.0	0.0
5	Useful heat from fuel Items No. 1 minus 2 + 3 + 4	1,228,300	80.66
6	Heat imparted to the air based on volume as read by Pitot tube and air temperature rise	1,232,000	80.8

The test procedure was to allow the heater to warm up for at least one hour, then take data at 15-min intervals for a period of 2 hours or more, for each test.

The results of the tests on the larger heater rated at 1,000,000 Btu per hour output capacity, fired with natural gas, are shown in the accompanying set of curves, Fig. 8, which represent the average of the many tests corresponding to the plotted points.

Table 1 gives the data and heat balance for a test at approximately 1,000,000 Btu output per hour. Table 2 is for a test at 1,250,000 Btu output per hour.

Discussion of Results of Tests

It will be noted that the motor horsepower required for 1,000,000 Btu per hour output was the 5 hp aimed at in the design. It will also be noted that the thermal efficiencies, even at 30 per cent overload, are all above 80 per cent. When firing natural gas in any apparatus having heat-transmitting walls of iron or steel, the gases must not be cooled to such a temperature that condensation will occur, otherwise rapid destruction by corrosion will occur. In view of the fact that the loss due to the latent heat of the water vapor formed by the combustion of natural gas amounts to about 10 per cent of the higher, or gross calorific value, the efficiency referred to the latter value could not, even under ideal conditions, exceed 90 per cent with natural gas. The test efficiencies, therefore, are in reality of the order of 90 per cent of this maximum attainable value.

After completion of the test in the Pittsburgh laboratory, a standard 1,000,000 Btu heater was shipped to Cleveland and tested for *AGA* approval. In these tests, safety is the main consideration. No heater of such large capacity had previously been tested in the *AGA* laboratories, but the tests were successfully completed and the unit when equipped with controls described bears *AGA* approval.

The temperature control of the gas-fired unit is of the on-and-off type, actuated by a *thermostat* located on the panel board. When the temperature of the surrounding air falls below the setting, the thermostat operates a contactor which starts the motor driving the fans. There is then a purging period of 30 sec during which time the exhauster fan removes any accumulation of gas from the combustion chamber. At the end of this time the *pilot valve* opens and the electric spark ignition comes on simultaneously to light the pilot gas. The gas pilot flame warms a thermocouple (of pilotstat) which makes the electrical contact to open the main gas valve. Approximately one minute is required, after the pilot lights, to bring the thermocouple up to temperature and cause the pilotstat to open the main gas valve. One minute later the electric spark ignition is cut off and the heater in then in normal operation.

If for any reason the pilot does not light, the entire heater is shut off 45 sec after the pilot valve opens. The safety device requires manual re-setting before the cycle can be repeated. If the pilot fails and goes out at any time during the operation, the pilotstat breaks the circuit and the main gas valve closes in about 30 sec.

If the gas pressure should drop to less than about one-half the normal value, a *vaporstat* in the gas line between the gas regulator and the burner

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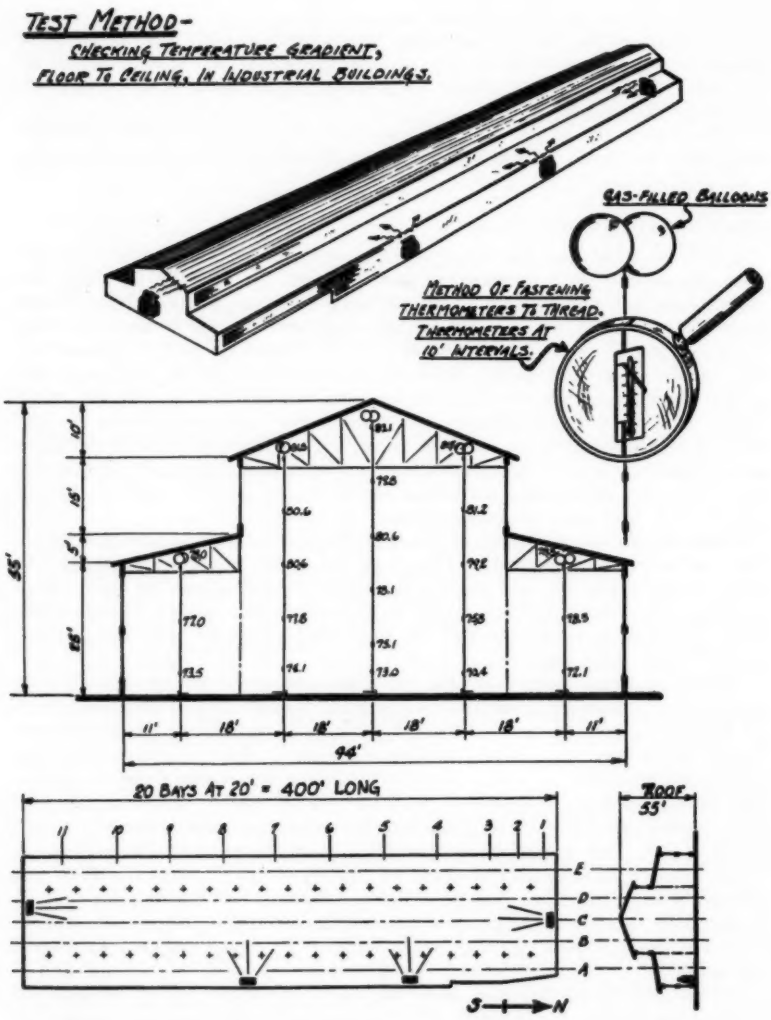

FIG. 9. SKETCH SHOWING LOCATION OF TEST STATIONS IN AN INDUSTRIAL BUILDING, AND AVERAGE TEMPERATURES AT VARIOUS LOCATIONS

cuts out the main gas valve. This instrument also requires manual re-setting before the cycle can be repeated.

A *sail-switch* is located in the air stream. If for any reason, such as burned-out motors or broken belts, the fans are not creating sufficient air velocity to throw this switch, the main gas valve cannot open.

A *temperature limit switch* is placed near the air discharge louvers. When the discharge air temperature reaches a pre-determined maximum, this switch causes the main gas valve to close. A *pressure switch* (sensitive vaporstat) in the stack also cuts out the main gas valve in the event the stack should become clogged; an increase of pressure in the stack equal to ½ in. water column is sufficient to actuate this switch.

After the room is up to temperature, the thermostat cuts off the gas to the burner. A *furnacestat* is provided to permit the fans to operate until the unit is cooled, after which time the fans are automatically turned off. When the thermostat again calls for heat, this cycle is repeated.

THE ENGINEERING OF THE APPLICATION OF UNIT HEATERS TO BUILDINGS

In order to decide upon the most suitable application of unit heaters to a given building, it is necessary to determine, *first*, the heat loss from the building; *second*, the number of units to be used; *third*, the location of the units with regard to the plan of the building and with regard to height.

The heat loss from the building is of course determined by the standard methods as stated in THE ASHVE GUIDE. However, it will not do to assume that the temperature at the walls and especially that under the roof are the same as the desired temperature either at the floor level or at the operators' working or breathing level. Experience seemed to indicate that for a given temperature at the working level, approximately 6 ft from the floor, the temperature under the roof for example was likely to be considerably lower when unit heaters, especially the floor mounted type, were used than when suspended heaters or those without forced air circulation are used. For direct radiation heating, the heat losses can be predetermined quite accurately by the use of factors published in THE ASHVE GUIDE and similar manuals, but no publication known to the authors included the data desired, applying when floor mounted, forced circulation, direct fired-heaters are used.

Accordingly, it was decided to make temperature measurements in an actual industrial building of large size. This machine shop building is 400 ft long by 94 ft wide by 55 ft high from the floor to the peak of the roof as shown in Fig. 9; 100-ft length of the building at the south end has a floor level approximately 15 ft higher than the remaining portion of the building, and a space about 80 ft long by 20 ft wide at the southwest corner is enclosed and could not be entered during the tests. One gas-fired heater is located in the middle of each end of the building, and two heaters are located at approximately even spacing along the 400-ft east wall. Each of these floor-mounted heaters delivers 12,000 cfm or a total of 2,880,000 cu ft per hour; the volume of the building is approximately 1,343,000 cu ft, and in terms of building volume, the air is turned over a little more than two times hourly. This factor is largely responsible for reducing the temperature differentials between floor and ceiling. The heaters drawing air from the floor line deliver it to the room at a velocity of about 2000 fpm, with an effective air carry

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across the entire width of the building. The distance carry is determined by the laws governing the dissipation of velocity of jets, which laws however are not very generally known. Among the factors affecting the carry are the size of the air stream, its initial velocity, its direction (whether having an upward or a downward component), its temperature differential above the surrounding air, and the presence of obstructions, which may either block the stream or create eddy currents which cut down the effective throw. A low roof on the other hand may increase the throw over that in a high building. In our test, the two middle nozzles on the heaters were generally pointed toward the middle of the building or at right angle to the side walls, with

TABLE 3—READINGS OF TEMPERATURE IN AN INDUSTRIAL BUILDING
(*Refer to Fig. 9 for location of measuring stations*)

Cross Section	Station					Location of Thermometers Below Roof	Cross Section	Station					Location of Thermometers Below Roof
	A	B	C	D	E			A	B	C	D	E	
1	79	80	80	78	78	1st below roof	6	80	81	84	80	79	1st below roof
	..	80	80	78	..	2nd below roof		..	82	80	81	78	2nd below roof
	..	80	80	76	..	3rd below roof		..	80	82	78	..	3rd below roof
	..	79	77	80	..	4th below roof		..	78	80	78	..	4th below roof
	..	..	76	..	..	5th below roof		..	..	74	..	..	5th below roof
	69	70	72	72	65	Breathing level		78	72	72	70	74	Breathing level
3	74	82	82	80	79	1st below roof	7	80	82	83	80	78	1st below roof
	..	82	78	78	..	2nd below roof		..	80	80	80	78	2nd below roof
	..	79	82	87	..	3rd below roof		..	80	81	79	..	3rd below roof
	..	74	79	82	..	4th below roof		..	78	79	73	..	4th below roof
	..	..	74	..	..	5th below roof		..	..	75	..	..	5th below roof
	68	64	68	66	70	Breathing level		79	72	70	70	73	Breathing level
5	81	82	82	80	76	1st below roof	9	..	80	85	84	82	1st below roof
	..	80	78	80	76	2nd below roof		72	80	82	..	80	2nd below roof
	..	80	80	80	..	3rd below roof		..	75	80	80	..	3rd below roof
	..	78	78	78	..	4th below roof		..	..	78	75	..	4th below roof
	..	..	76	..	..	5th below roof		..	..	74	..	..	5th below roof
	81	79	74	70	70	Breathing level		74	74	72	70	74	Breathing level

Note: All temperatures are in degrees Fahrenheit. Outside temperature at time of test was 36 F. First thermometer was approximately 2 ft below roof; second was 10 ft below first; third was 10 ft below second, etc., at 10 ft intervals. Breathing level was 5 ft above floor.

the two end nozzles at a slight angle (approximately 30 deg) away from the direction of discharge from the central nozzles.

For making the test, some means had to be provided for locating temperature sensitive elements at the various levels in the building. Of the various ideas suggested, such as telescopic means, towers, pulleys and strings, and balloons, after eliminating those that were costly or that did not permit of immediate testing, the balloon method was decided upon. Light thread was fastened to hydrogen filled balloons, and thermometers were fastened to the string at a point just below the balloons, and at 10 ft intervals to the floor and at the breathing level about 5 ft from the floor.

Five stations were located across the width of the building, approximately 20 ft apart as shown in Fig. 9, with balloons and string of thermometers at each. Eleven stations were located along the length of the building, the first 3 being at the middle of the 20-ft bays beginning at the north wall, and the remaining 8 being at two-bay or 40-ft intervals.

The procedure was to record the temperature at one station, then move to the next and do the same, until all stations in one cross-section of the building had been covered; then move to the next cross-section along the length, read and record all temperatures at successive stations in this cross-section; and repeat until all cross-sections along the length had been covered. At each station, after allowing time for the thermometers to reach temperature equilibrium, one man pulled the balloons down and read the thermometers in rapid succession while another man recorded the temperatures. To make sure that the thermometer readings did not change while being pulled down, in the

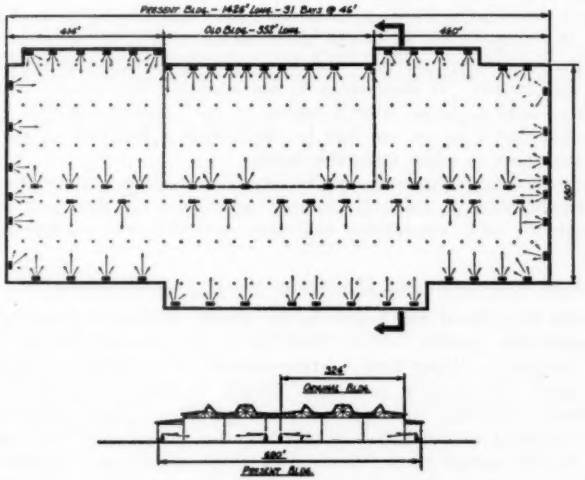

FIG. 10. PLAN OF ORDNANCE PLANT, SHOWING LOCATION OF HEATERS

short time between leaving their positions and being read, a test of the time lag of the thermometers had previously been made. All of the thermometers on one suspended thread could be read in approximately 15 sec, and the time lag of the thermometers was so much greater than this that the change in the reading while lowering was negligible.

The hydrogen-filled balloons supporting the threads were about 15 in. in diameter when inflated. With six thermometers attached, two balloons were required per thread or per station, as one balloon was not sufficient to support them; however, with two balloons attached, if one burst from hitting some sharp projection just below the roof, the other furnished sufficient buoyancy and resistance to falling, to give plenty of time to get to this particular station and catch the thermometers before they reached the floor.

The temperature marked at each location in Fig. 9 is the average of the readings at the various points along the length of the building. In Table 3 are given the tabulated measurements, from which the variations along the length of the building, and the influence on such variations of proximity to

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or distance from the heaters, can be judged. The outside temperature at the time of these tests was about 36 F. It will be noted that from the breathing level up to a point just below the peak of the roof the increase of temperature averaged 10.1 F. This gives a basis upon which the heat losses from a building heated by floor mounted unit heaters can be calculated.

The number of heaters required is then calculated from the capacity of a unit, which ranges from 750,000 to 1,500,000 Btu per hour. As to location, floor mounting is preferred wherever it is feasible; and while the heaters can either be located along the walls discharging inward toward the middle of a wide building, or near the middle discharging outward toward the walls, the former is preferred. Although no exact comparative tests have been made, it is judged that placing the heaters in the middle discharging toward the walls increases the temperature along the walls, where it will do no good, but will increase the heat loss; and also that it results in slightly less uniform temperature distribution. If floor space is not available, the heaters can be suspended from side walls or from columns or roof trusses; in this case it is recommended that a return air duct be run within a few feet of the floor so that the colder air is taken from this level.

Locating the heaters near the walls simplifies venting. Since the unit includes an exhauster, no stack is required, and often the flue need merely be run through the wall, extended a few feet vertically and weather-capped.

The Relation of Direct-Fired Heaters to the Defense Program

The large direct-fired unit heater in the present industrial situation assumes added importance because: *First,* heat for large projects can be furnished quickly. *Second,* a direct-fired heating system can readily be expanded, as building extensions become necessary, simply by increasing the number of units, without requiring expansion of a central plant. *Third,* not only is less labor and material required than in other heating systems, but the engineering required is less complicated, which is a great advantage at present when engineers are scarce. The following example illustrates these advantages and their relationship to national defense.

A large machine shop, built for the U. S. Naval Ordnance, was heated by twenty-two 1,000,000 Btu gas-fired units (Fig. 10). The shop was enlarged to an area about six times the original plan with a total hourly heat loss of about 80,000,000 Btu. The heating plant was easily expanded to meet the new condition by simply adding 58 more units.

Thus the advent of the direct-fired unit heater adapted to either gas or oil firing is particularly timely with reference to the national defense program.

DISCUSSION

E. K. CAMPBELL, Kansas City, Mo.: You referred to the fact that the output of a heater is approximately 6000 Btu per square foot. Do you recommend that this rated capacity be operated at all times, or is there a limiting output that you recommend from the standpoint of durability? I would also like to inquire what the operating outlet temperature of the unit is, or the temperature rise of the air as it passes through the unit.

W. M. MYLER, JR., Columbus, Ohio: Before asking a couple of questions I would like to state that there are both steel and cast-iron units available for the type of service referred to by the authors of this paper. In studying the tables of data that were presented, there were a few things that seemed a little unusual for gas-fired equipment, and that is the basis on which these questions are asked.

The carbon dioxide values shown in these tests are abnormally high as compared to what we normally consider for gas-fired equipment. For instances, values are given around 10.2 and over 11 per cent, with one case showing an oxygen content of less than 2 per cent. How is this high carbon dioxide controlled and is it a field or factory adjustment?

Two tests are shown operating at a 25 per cent difference in rating. Normally a change in rating of this amount would cause a corresponding change in the carbon dioxide and yet we did not find anything reported that would indicate such an alteration. Does this mean a change in construction of the unit or simply a change in adjustment of the heater between the two test conditions? A reference was made to the *American Gas Association* approval which apparently applies only to the million Btu unit, but I would like to inquire whether it also applies to the higher rating.

Do the carbon dioxide values, which are much higher than we normally find in the *AGA* approved appliances, actually correspond to the figures of that same unit when it was tested and received *AGA* approval. If the answer to this question is yes, then there is another question that comes to my mind. How do you take care of the normal fluctuations in Btu content, because we know in some localities variations of natural gas exist which have extremes as high as from 1025 to 1175. Obviously, if the heater is adjusted to a gas heat content of 1025 with a 2.1 per cent carbon dioxide, it would seem trouble might develop if gas having a heat content of 1175 is introduced.

F. E. SPURNEY, Washington, D. C.: I would like to ask a question about the maintenance of these heaters. What is the longest period of time you have had such units installed and how did they stand up in actual practice? What are the maintenance costs?

L. E. SEELEY, New Haven, Conn.: I am wondering whether the 10 F temperature gradient mentioned would apply to all outside temperatures, or was there some particular temperature at which this was determined.

MR. PIETSCH: With reference to Mr. Campbell's question, the output on the basis of 6,000 Btu per square foot is considered an average. In the upper half of the unit most of the heat transfer is by radiation, but in the lower half, which is covered by plastic material, some of the heat released is obviously obtained by a combination of conduction and radiation. Hence, the heat transfer is not as high in the lower section as it is in the upper half of the heater. We have estimated from test results that from 75 to 80 per cent of the heat is transferred in the upper half of the heater by radiation and a small amount by convection. In order to maintain the proper combustion chamber temperatures we have found that the air velocities over the heating surface run approximately 45 fps. The outlet temperature rise of the heater depends on the amount of air circulated but it runs approximately 70 to 80 F above room temperature.

The high carbon dioxide obtained can only be produced with controlled combustion, which is accomplished by two means. At the gas burner, combustion air can be controlled from the supply fans, can also be controlled by a damper in connection with the exhaust fans, and of course, by the control of the gas supply. The burner is a pre-mix type and consequently most of the air is mixed before it is fed to the burner.

The 25 per cent difference in rating is accomplished by increasing the volume of the fans. The heat transfer is increased as a result of the extra air velocity and we maintain the proper carbon dioxide by controlled combustion. The difference in 25 per cent of the rating is not accounted for exactly by a difference in construction.

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The million and the million and a quarter Btu units are identical with exception of the amount of gas burned and the amount of air circulated over the heater.

Difference in Btu content of gas in various localities is taken care of by the proper selection of burners so that we obtain within a few cubic feet of gas, the required volume that is necessary for a particular installation. Service engineers are assigned to each installation to set the performance for the desired local condition. Obviously, high carbon dioxide maintained on test is not held in the field where we set the combustion rate around 9 to 10 per cent which will allow for a certain fluctuation of the Btu value of the gas. The tests of the *AGA* do not operate as high as the carbon dioxide mentioned in the paper, due to the fact that it is not necessary in their tests to carry such a high carbon dioxide value. Under these conditions the carbon dioxide ran approximately 9 to 10 per cent on the equipment in the test laboratory.

It is important to keep the combustion chamber cool and below the point at which the metal will oxidize. We have combustion chamber temperatures ranging from 1600 to 1800 F, but the metal temperature is always below 700 F.

Regarding the temperature gradient that can be maintained between the floor level and the ceiling the data reported in the paper were obtained when the outside temperature was 36 F. I cannot say whether it would be any different for a lower outside temperature, but I do not imagine it would be much greater.

No. 1203

CORROSION TESTS IN A WATER-RECIRCULATING AIR CONDITIONING SYSTEM

By W. Z. Friend,* New York, N. Y.

This paper results from an investigation sponsored by the ASHVE Research Technical Advisory Committee on Corrosion.†

IN water recirculating types of air conditioning systems, the problem of corrosion has been considerably moderated by the common use of chemical treatment of the wash water. So far as the dehumidifier or water spray chamber itself is concerned, corrosion can be virtually eliminated by the use of proper chemical treatment and this protection seems to extend to all parts of the chamber wetted with the water spray and to the prevention of serious galvanic action between non-ferrous valves or spray nozzles and adjoining steel or galvanized piping.[1] However, this chemical protection does not extend to the portions of the equipment directly beyond the eliminators. Consequently, in these parts—the fan and fan housing, ducts immediately adjacent to fan housing, duct between fan and eliminators, and parts of the eliminators themselves—corrosion may still be a serious problem.

Corrosion in the previously described parts of air conditioning systems is usually due to the presence of sulfur dioxide in the entering air, which when mixed with water forms dilute sulfurous acid. As would be expected, corrosion is usually most severe in industrial districts or in large cities where the sulfur dioxide content of the air is highest.

Treatment of the wash water with the proper combination of sodium dichromate and sodium hydroxide is usually effective in preventing appreciable corrosion in the dehumidifier itself by neutralization of part of the sulfurous acid and inhibiting attack by the remainder. However, the sulfur dioxide is usually not entirely removed from the air by this treatment[2] and, furthermore, the dichromate is not carried over into the air stream leaving the spray chamber.[3] Alternate changes in temperature of the air stream caused by the operation of automatic controls result in the condensation and evaporation on metal surfaces of aerated water. This water condensing from the vapor phase

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[1] The Control of Corrosion in Air Conditioning Equipment by Chemical Methods, by C. M. Sterne. (*ASTM Proceedings*, Vol. 35, Part II, 1935.)

[2] A Study of the Removal of Sulfur Dioxide from Library Air. (*National Bureau of Standards*, Miscellaneous Publication No. 142.)

[3] Loc. Cit. Note 1.

Presented at the 48th Annual Meeting of the American Society of Heating and Ventilating Engineers, Philadelphia, Pa., January, 1942.

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in the air stream contains no inhibitor and may contain traces of sulfur dioxide so that it is highly active as a corroding agent. Consequently, it is to be expected that corrosive conditions will be most severe in the parts of the system just beyond the eliminators.

The tests described in this paper were made to determine the resistance of a number of common ferrous and non-ferrous metals and alloys to corrosion under the conditions existing beyond the eliminators, and also in the wash

TABLE 1—ANALYSES OF WASH WATER—PARTS PER MILLION
(*Water filtered before analysis*)

ANALYSIS NO.	1	2	3	4
Date	1-2-39	1-3-39	5-24-39	5-24-39
LOCATION	NORTH DEHUMIDIFIER	SOUTH DEHUMIDIFIER	NORTH DEHUMIDIFIER	SOUTH DEHUMIDIFIER
MINERAL SUBSTANCES				
Silica	12	10	4	5
Iron and Alumina	13	13	2	2
Calcium Oxide	55	55	2	8
Magnesium Oxide	3	3	...	...
Bicarbonates as Calcium Carbonate	175	180	160	190
Chlorides	28	28	18	18
Sulfates	406	449	102	92
Chromates as Sodium Bichromate	452	365	313	313
Suspended Matter	Present	Present	Present	Present
Total Solids (Gr.-Gal.)	83.5	83.1	40	40.4
Hardness as $CaCO_3$ (Gr.-Gal.)	5.9	5.9	0.2	0.8
Total Incrusting Solids (Gr.-Gal.)	11.3	11.3	0.7	1.8
Total Non-Incrusting Solids (Gr.-Gal.)	72.2	71.8	39.3	38.6
Organic Matter	Present	Present	Present	Present
pH	7.9	7.9	7.9	7.9

water inside the dehumidifier in a system where proper chemical treatment was being given. The air conditioning system of the National Broadcasting Co. at New York was selected as the location for the tests because it is representative of best modern practice, handles a large volume of large-city air, operates 20 to 21 hours per day during the entire year, and because of the excellent cooperation and supervision provided by the operating personnel. The operations so far as air washing is concerned follow the usual sequence. Air is drawn in from tenth-floor level through dry filters and baffles into the water spray chambers, then through the eliminators, the fan, and out into the duct system. The wash water is recirculated and in summer is cooled by refrigeration. Typical analysis of the circulating wash water taken from both the north and south dehumidifiers are given in Table 1. These show the difference in the water when the equipment is running during the refrigeration season (dehumidifying) and during the winter season (humidifying). All four analyses were made during the corrosion test period.

During the refrigeration season, water is being removed from the air and there is a certain amount of water running to waste through the overflow line at all times. Consequently, the water is being diluted with what is in effect distilled water. This condition is shown by analyses 3 and 4. During the

winter season, water is being added to the air at all times. There is no loss of water by means of overflow, but there is a constant inflow of water to make up for that evaporated into the air stream. Hence, there is a concentrating effect in the wash water as shown by analyses 1 and 2.

All of the recirculated wash water is treated with both sodium dichromate and sodium hydroxide. The dichromate concentration is maintained at 300 to 350 ppm and the *pH* value at 7.8 to 8.2. Determinations of *pH* are recorded continuously in a glass electrode recording potentiometer. Sodium dichromate

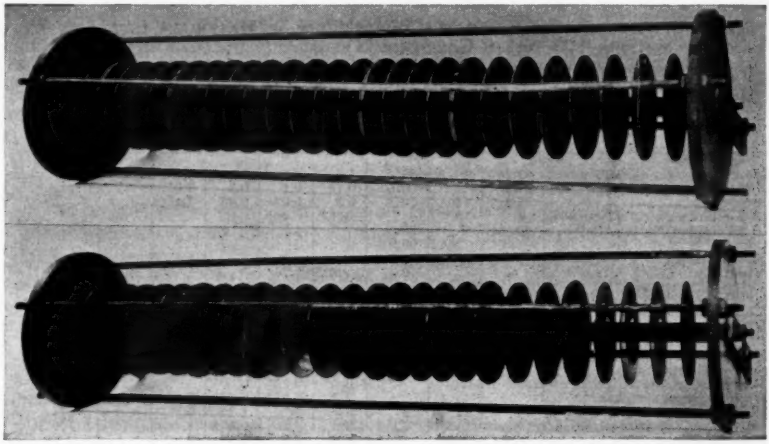

FIG. 1. CORROSION TEST SPOOLS AFTER REMOVAL FROM TEST AND BEFORE CLEANING

Top: Spool from sodium dichromate—sodium hydroxide treated water in dehumidifier
Bottom: Spool from washed air stream between eliminators and fan

determinations in each recirculating water circuit are made daily by the potassium iodide-sodium thiosulfate titration method.

The corrosion test specimens were assembled on test spools [4] as shown in Fig. 1. The method of testing was in accord with the *ASTM* Tentative Recommended Practice for Conducting Corrosion Tests.[5] Briefly, the method consists in assembling uniform, weighed specimens of the various materials on a spool so that they are completely insulated from each other and from the metallic parts of the spool. A plastic tube covers the center structural rod of the spool and the specimens (each having a hole in the center) are strung on this tube with pieces of larger plastic tube 5/8 in. long between each specimen. On each end of the spool is a 4 in. diameter plastic disc which serves to keep the specimens away from the sides of the operating equipment and to act as support for other structural rods.

Typical analyses of metals and alloys included in these tests are given in Table 2. The test specimens of wrought materials are made from 1/32 in.

[4] Corrosion Testing Methods, by H. E. Searle and F. L. LaQue. (*ASTM Proceedings*, Vol. 35, Part II, 1935.)
[5] Tentative Recommended Practice for Conducting Plant Corrosion Tests (*ASTM* Designation: A224-39T, 1939).

sheet or strip of the commercial materials. They are machined to exactly 2.233 in. outside diameter with a 21/64 in. diameter hole in the center. These dimensions are chosen so that when assembled on the spool the exposed area of each specimen is 0.5 square decimeter. Specimens of cast materials were machined to 3/16 in. thickness with the diameter correspondingly smaller so that exposed area was the same as for wrought materials. Before assembly, the specimens were prepared by scrubbing with powdered pumice and a bristle brush to remove dirt or rust completely, rinsing in water followed by alcohol-ether, drying, and weighing on an analytical balance to the nearest 0.0001 gram.

TABLE 2—APPROXIMATE CHEMICAL COMPOSITIONS OF MATERIALS INCLUDED IN CORROSION TESTS

MATERIAL	APPROXIMATE CHEMICAL COMPOSITION—PER CENT BY WEIGHT								
	Ni	Cr	Cu	Fe	C	Mn	Si	Zn	Other
Monel	67.	...	30.	1.4	0.15	1.	0.1	...	...
Nickel	99.4[a]	...	0.1	0.15	0.1	0.2	0.05	...	...
Inconel	79.5	13.	0.2	6.5	0.08	0.25	0.25	...	...
18–8 Stainless Steel (Type 304)	8.	18.	...	Bal.*	0.08[b]	...	...	...	...
70–30 Copper-Nickel Alloy	30.	...	70.	...	...	...	...	...	...
Copper	...	...	99.9 +	...	...	...	...	...	...
Silicon Copper	...	...	94.5	...	...	1.	4.5	...	...
Admiralty	...	...	70.	...	...	...	...	29.	Sn — 1.
Zinc	...	...	...	0.05	...	...	...	98.65	{Pb — 0.8 / Cd — 0.5}
5% Nickel Steel (SAE 2512)	5.	...	...	Bal.	0.20[b]	0.60[b]	...	...	...
3½ Nickel Steel (SAE 2315)	3.5	...	...	Bal.	0.20[b]	0.60[b]	...	...	...
Nickel-Copper Alloy Steel	2.	...	1.	Bal.	0.08	0.44	...	...	...
Copper-Bearing Steel	...	...	0.20[a]	Bal.	0.07	0.35	...	...	...
Mild Steel	...	...	...	Bal.	0.25[b]	0.60[b]	...	...	...
Wrought Iron	...	...	...	Bal.	0.05[b]	0.05[b]	0.15[b]	...	...
Ni-Resist	14.	2.	6.	Bal.	3.	1.25	1.5	...	...
3% Nickel Cast Iron	3.	...	...	Bal.	3.2	0.5	1.	...	...
Plain Cast Iron	...	...	...	Bal.	3.2	0.5	1.5	...	...

[a] Including cobalt. [b] Maximum. [c] Minimum.
* *Balance* in every case includes impurities not otherwise listed.

Four spools were prepared—two for each test location. Each spool contained one specimen of each material being tested. This gave duplicate specimens in each location.

The spools were placed in test in the north unit on February 22nd, 1938, while the equipment was shut down for a short period at night. Two of the spools were suspended in the wash water in the bottom of the spray chamber not far from the water overflow. They were suspended horizontally so that the specimens were always immersed in water but above any layer of solids which might collect on the bottom of the chamber. The other two spools were suspended horizontally by wires in the washed air stream between the eliminators and the fan. They were located about 24 in. from the eliminators and about centrally from side to side and top to bottom.

The tests were discontinued on October 5, 1939, giving a test period of 590 days. The spools were taken directly from the equipment, allowed to dry and photographed. One spool from each location is shown in Fig. 1. The specimens were removed and cleaned by electrolytic treatment in an inhibited acid bath,[6] followed by brushing, washing in water and in alcohol-ether, drying and weighing. Corrosion rates were calculated in milligrams loss per square

[6] Loc. Cit. Note 5.

decimeter per day (Mdd) and inch penetration per year (Ipy) as shown in Tables 3 and 4. The rates shown are the average of two specimens of each material. In converting from milligrams per square decimeter per day to inch per year, the following equation was used:

$$\text{Ipy} = \text{Mdd} \times \frac{0.001437}{d}$$

where

d = density of metal or alloy in grams per cubic centimeter.

Depth of pitting or localized corrosion was measured microscopically with an accuracy of 0.001 in. Column 4 of Tables 3 and 4 shows the maximum depth of pitting, if any, which occurred at and in the crevice between the specimen and the adjoining plastic spacer. Column 3 shows the maximum depth of pitting, if any, which occurred on the exposed surface of the specimen.

RESULTS IN WASH WATER

The weight losses of all the materials tested in the dichromate treated wash water were extremely low and confirm the effectiveness of this treatment in inhibiting corrosion. The general performance of mild steel and plain cast-iron is so good that there would ordinarily be no point in using more expensive materials for the fabrication of the spray chamber itself where this treatment is given. However, particular attention should be given to the pitting which occurred underneath and adjacent to the spacers with all of the steels and cast irons tested. Practically all of the *surface* pitting shown in Column 3 of Table 3 actually occurred in areas adjacent to the spacers where solid deposits from the water might get a foothold in the crevice between specimen and spacer and tend to accumulate. This pitting is similar to that which may

TABLE 3—CORROSION TESTS IN RECIRCULATED WASH WATER IN DEHUMIDIFIER

MATERIAL	AVG. CORROSION RATE		MAX. DEPTH PITTING—INCH	
	Mdd	Ipy	Surface	Under Spacer
Monel	0.005	<0.00001	None	None
Nickel	0.013	<0.00001	None	None
Inconel	0.006	<0.00001	None	0.002
18–8 Stainless Steel (Type 304)	0.001	<0.00001	None	0.001
70–30 Copper-Nickel Alloy	0.007	<0.00001	None	None
Copper	0.011	<0.00001	None	None
Silicon Copper	0.024	<0.00001	None	None
Admiralty	0.024	<0.00001	None	None
Zinc	0.27	0.00006	0.012	0.009
5% Nickel Steel (SAE 2512)	0.21	0.00004	None	0.007
3% Nickel Steel (SAE 2315)	0.28	0.00005	0.006	0.010
Nickel-Copper Alloy Steel	0.93	0.00017	0.006	0.010
Copper-Bearing Steel	0.22	0.00004	0.004	0.007
Mild Steel	0.25	0.00005	0.005	0.007
Wrought Iron	0.21	0.00004	0.003	0.012
Ni-Resist	0.20	0.00004	0.003	0.011
3% Nickel Cast Iron	0.56	0.00011	0.005	0.009
Plain Cast Iron	0.54	0.00011	0.004	0.017

< = less than.

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TABLE 4—CORROSION TESTS IN WASHED AIR STREAM BETWEEN ELIMINATORS AND FAN

MATERIAL	AVG. CORROSION RATE		MAX. DEPTH PITTING INCH	
	Mdd	Ipy	Surface	Under Spacer
Monel	0.19	0.00003	0.001	0.001
Nickel	0.09	0.00001	0.002	0.003
Inconel	0.015	<0.00001	0.004	0.003
18–8 Stainless Steel (Type 304)	0.015	<0.00001	0.001	0.003
70–30 Copper-Nickel Alloy	0.42	0.00007	None	None
Copper	0.49	0.00008	None	None
Silicon Copper	0.42	0.00007	None	None
Admiralty	0.59	0.00010	None	None
Zinc	0.23	0.00005	Incipient	None
5% Nickel Steel (SAE 2512)	24.1	0.0044	*	*
3% Nickel Steel (SAE 2315)	35.7	0.0065	*	*
Nickel-Copper Alloy Steel	28.2	0.0052	*	0.006
Copper-Bearing Steel	36.9	0.0068	*	0.006
Mild Steel	54.2	0.010	Perforated	*
Wrought Iron	37.4	0.0069	Perforated	0.004
Ni-Resist	6.4	0.0012	*	0.006
3% Nickel Cast Iron	46.3	0.0091	*	0.005
Plain Cast Iron	62.1	0.0124	*	*

< = less than. * Roughened surface due to general corrosion.

occur with some metals and alloys due to the presence of oxygen concentration cells.[7] In that case the surface of metal freely exposed to an aerated solution is passivated by the oxygen present. However, in crevices or underneath surface deposits not receiving free circulation and aeration, the oxygen may become used up and the metal lose its passivity and assume an *active* state. Under these conditions a small electrolytic cell may be set up between the passive and active surfaces causing accelerated corrosion of the latter. Under favorable conditions, cells of this type may develop potentials as high as 500 millivolts or more.

In the present case, potassium dichromate is a *passivator* of steel and its corrosion inhibiting properties are due to this fact. It is probable that in this test the accumulation of some solid deposit from the water at the spacers prevented the free access of dichromate to the surface in and near the crevices with the result that these surfaces were in a more *active* condition than the adjoining surfaces and thereby suffered accelerated corrosion of the concentration cell type. In the case of mild steel, the maximum depth of this pitting was 0.007 in. in 590 days. Where the metal used is fairly heavy, such a rate may not be particularly serious. Also, the pitting may tend to be more prevalent in the wash water in the bottom of the spray chamber where solids tend to accumulate than it would be on the sides of the chamber which are continuously washed down. However, it is apparent that cracks and crevices should be avoided as much as possible in fabrication. The pitting which occurred in the zinc specimen is of particular significance since zinc is generally applied as a galvanized coating which would soon be penetrated by this rate of localized attack.

[7] Corrosion Resistance of Metals and Alloys, by R. J. McKay and R. Worthington. (*American Chemical Society*, Monograph No. 71, Reinhold Publishing Corp., 1936.)

RESULTS IN AIR WASHED STREAM

The results of the tests in the washed air stream are given in Table 4. A further idea of the severity of corrosion may be obtained from Fig. 2 showing one cleaned specimen of mild steel from each test location. In the air stream tests, as in many atmospheric exposures, the alloy steels show definite advantages over mild steel, 5 per cent nickel steel and nickel-copper alloy steel, having twice the resistance of mild carbon steel. Corrosion of all the steels is sufficiently great, however, that where an unprotected metal surface is to be used, the selection would probably be made among the high nickel or copper alloys, stainless steels, or possibly galvanized steel. The zinc specimens showed evidence of some incipient pitting (numerous very fine pits less than 0.001 in. deep) and, hence, would be questionable for this location between eliminators

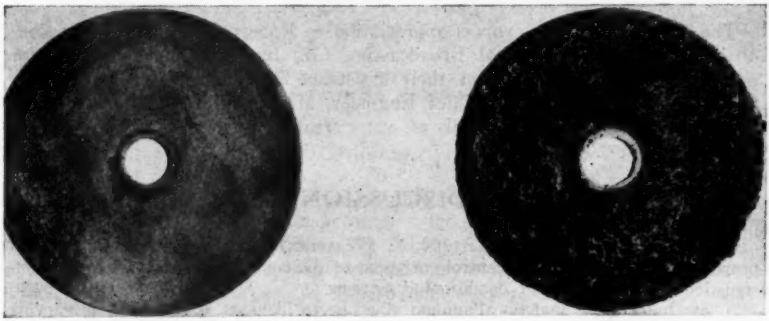

FIG. 2. MILD STEEL SPECIMENS AFTER REMOVAL FROM TEST AND CLEANING
Left: Specimen from sodium dichromate—sodium hydroxide treated water in dehumidifier. Note local attack adjacent to spacer. *Right:* Specimen from washed air stream between eliminators and fan.

and fan where condensation may occur. In the dryer portion of the duct system beyond the fan, galvanized steel is widely used and is apparently giving good performance.

SUMMARY

In actual practice the use of mild steel and plain cast-iron covered with a suitably resistant asphalt-base paint is giving good performance in the spray chambers and eliminators of dichromate-treated systems which are readily accessible for frequent inspection and repair. The coating tends to fill in any cracks or crevices of the equipment and round out sharp corners where solids might otherwise collect. It also provides protection against corrosion in case the chemical treatment is discontinued for short periods due to neglect or breakdown. In a system where frequent inspection and repair cannot be made it is conceivable that the coating might, under certain conditions, do more harm than good. Cracks or holes in the coating which persisted for a considerable time might provide the starting point for accelerated localized corrosion. In any event, it is advisable to avoid cracks and crevices as much as possible. In the spray chamber in which these tests were made, unpainted

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nickel-copper alloy spray nozzles are being used with painted steel nipples without appreciable galvanic corrosion of the latter.

The most severe corrosive condition exists in the washed air stream from the eliminators to the fan. In some systems the use of acid-resisting synthetic resin-base paints is apparently giving satisfactory protection in this area. Nickel or nickel-copper alloy steels have sufficiently better resistance than mild steel to warrant their use under these paints for protection in case the coating becomes broken or worn off for considerable periods without repair. For parts not available for frequent inspection and repair the use of the highly resistant metals or alloys, such as the high nickel or copper alloys or stainless steels, is indicated. This applies particularly to the fan and fan housing which may be subject to considerable erosion as well as to corrosive attack.

ACKNOWLEDGMENTS

The author wishes to express appreciation to Robert Close, Chief Air Conditioning Engineer, National Broadcasting Co., Inc., and his associates for permission to conduct and for their assistance in conducting the foregoing tests; also, to C. M. Sterne, Chief Engineer, Metropolitan Refining Co., Inc., for water analyses and for details of water treatment.

DISCUSSION

C. M. STERNE, Long Island City, N. Y. (WRITTEN): It is a well known fact that proper water treatment does control corrosion of the common metals used in the construction of commercial air conditioning systems.

It was unfortunate that no aluminum alloys were included in the tests, as varying degrees of resistance to corrosion of these metals have been found under the test conditions outlined. Tests of various protective coatings should be made in this equipment, as the same conditions still prevail.

Fig. 2 is of very great interest, as the contact corrosion which appears adjacent to the spacer on the mild steel specimen from the treated water is now no longer a problem. By means of proper non-corrosive, neutral fungicides it has been found possible to completely inhibit all organic growth in the recirculated water. Formerly contact corrosion was a serious problem and some work was done by Mears and Evans in England in 1934, but little explanation was given.

It is the writer's opinion that contact corrosion is due to the formation of differential oxygen concentration cells in the crevice spaces at the point of contact due to the fact that organic matter can collect there and keep the treated water from contact at these points. We have found, however, that by using pentachlorophenols or pentachlorophenates that we no longer have organic growths in crevices, and therefore there is no formation of differential oxygen concentration cells.

This paper clearly proves the corrosion problems that are found beyond the eliminators. This type of attack is especially severe in evaporative condensers and induced draft cooling towers. The writer has found in many cases heavy scaled formation on evaporative condenser tubes and equally heavy corrosion of the fans and fan housings 2 ft away from the scaled portions of the equipment.

F. N. SPELLER, Pittsburgh, Pa.: The remarks of Mr. Sterne about fungus growth around the contact of the specimens should not be confused with what was previously mentioned about the general presence of organic matter in water. It is well known how algae and fungii can be stopped, but organic matter such as humus will reduce the chromates to a chromous salt, which has no productive value. This problem is not serious for city water, unless it has become contaminated, such as by drainage over roofing material or some other surface of this nature.

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No. 1204

ENTRAINMENT AND JET-PUMP ACTION OF AIR STREAMS

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Case School of Applied Science

SUMMARY

WHEN air is discharged from an outlet into a free-open space or room, the primary air stream becomes a *jet pump* for moving the air surrounding it. Determining the behavior of such an air stream when it is entraining and mixing the room air is a problem in fluid mechanics which must be solved largely by experimental methods, although approximations may be made by analytical means.

In view of the increasing variety of grilles, slots, nozzles and venturi devices used for diffusion of the air supply in a room, a brief general survey of the subject of air diffusion by jet devices is presented. The air conditioning engineer is interested in the jet pump not only in connection with air streams in rooms, but also for such applications as fume exhausters, gas and oil burners, smoke prevention jets and steam-ejector refrigeration. Hence this paper includes an examination of data and results reported by several investigators.

Experimental results are presented for air streams discharging into free open space from various types and sizes of outlets, at velocities up to 2400 fpm. Using an experimental method of integration, these results are shown to confirm the analysis by the theory of conservation of momentum. The total air entrained was in direct proportion to the distance from the discharge outlet, but the ratio of entrained room air to primary air was found to be almost independent of the velocity at the outlet face. For each foot of throw or travel of the stream, the actual volume of room air entrained was 0.3 to 0.7 times the primary air volume. This entrainment was increased by using a small outlet, a narrow slot, or a spreading grille.

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THE PROBLEM

The jet pump has a large number of applications, but each of these is so specialized that no comprehensive study of the device is available. The familiar puffing of a locomotive, the operation of the gas burner on a kitchen stove and the movement of air by a desk fan are common examples of air entrainment on the jet-pump principle.

Jet-pump equipment is built in a wide variety of forms, but for a general analysis Fig. 1 may be used to represent most of them. The essential part of the apparatus is the *nozzle* or discharge outlet from which the jet emerges. If the mixture is to be pumped or directed into an outlet pipe, the venturi or mixer section is used, but the entrainment or mixing action will take place even though the venturi element is entirely omitted and the jet discharges into

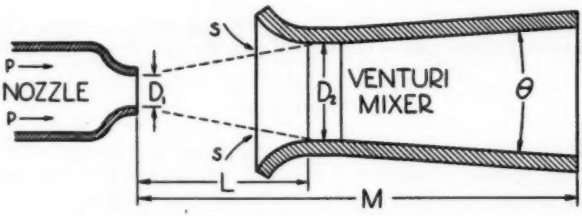

FIG. 1. DIAGRAM OF TYPICAL JET-PUMP APPARATUS

free space. The primary stream P may be any fluid, air, water, steam, gas, etc. The entrained or secondary fluid S is usually air or water, although any other fluid might be substituted.

The engineer is interested in two questions about the jet pump:

(1) What are the best design proportions or dimensions for the device?
(2) What are its performance characteristics?

PRINCIPLE OF JET PUMP

The essential feature of the jet pump is to obtain the maximum of velocity or kinetic energy at the nozzle discharge-face. For lower-pressure air or gas, and for all liquids, the process of accelerating the fluid as it approaches the nozzle discharge is by a simple conversion from pressure energy to kinetic energy:

$$\frac{P}{w} = \frac{V^2}{2g}, \text{ or } V = \sqrt{2gh} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where the velocity V in feet per second and the fluid density w in pounds per cubic foot represent the conditions at the nozzle discharge face. P is the drop in pressure, pounds per square foot, from the upstream section to the

nozzle face, or h is the equivalent head in feet of fluid flowing. As long as compressibility and velocity of approach are both negligible, this equation may be used to compute the velocity at the nozzle face for any fluid, gas, liquid or vapor. From the practical standpoint this process is merely that of increasing the velocity of flow by reducing the size of conduit. When the pressure-drop is large and the fluid is a gas or a vapor, the fluid expands as it approaches the nozzle outlet due to the reduction in pressure, and a still higher velocity is produced. With such a high-pressure nozzle, some of the internal energy of the fluid is also turned into kinetic energy and the calculation of velocity may be based on the drop in enthalpy for a constant-entropy process.

TABLE 1—DESIGN AND PERFORMANCE OF JET PUMPS FOR AIR ENTRAINMENT
(*See Fig.* 1)

APPLICATION	REFERENCE FOOTNOTE No.	FLUIDS		DESIGN							PRESSURES:		ENTRAINMENT RATIO
											JET	ACROSS SUPPLY MIXER	
		P	S	D_1	$\frac{D_2}{D_1}$	$\frac{L}{D_1}$	$\frac{M}{D_1}$	$\frac{L}{D_2}$	$\frac{M}{D_2}$	Angle θ deg	Psi	In. of Water	
Locomotive front-end	4	Steam	Gases	1.5	4.0	4	12	1	3	9	3.0	5	3.0
Fume exhauster	7	Air	Air	5.2	2.5	..	36	..	14	6	0.28	1.5	1.5
Fume exhauster	[a]	Air	Air	2.0	3.0	7	46	2.3	15	5	1.0	0	5.5
Fume exhauster	[a]	Air	Air	2.0	3.0	7	46	2.3	15	5	1.0	1.5	2
Gas burner	2	Air	Air	0.08	15	20	230	1.3	16	6	3.0	0	30
Gas burner	2	Air	Air	0.08	15	20	230	1.3	16	6	3.0	0.5	11
Gas burner	[b]	Nat. Gas	Air	..	..	..	..	2	8	4	..	..	..
Experimental	5	Air	Air	0.38	3.0	15	..	5	..	16	0.9	0.5	3
Smoke preventer	[c]	Steam	Air	0.12	12	25	120	1.2	6	7	80	3.0	15
Experimental	6	Air	Air	0.50	1.5	8	25	5	16	16	0.9	6.0	2

[a] The Venturi Ejector, by F. F. Kravath. (*Heating and Ventilating Magazine*, June 1940.)
[b] *U. S. Bureau of Standards Technologic Paper* No. 193, Design of Atmospheric Gas Burners, 1921.
[c] Design of Steam Jets for Smoke Abatement, by Du Perow and Bossart. (*Thesis*, Case School of Applied Science, 1927.)

But computing the nozzle velocity does not explain the pumping process. In what manner or by what means does this nozzle jet produce entrainment and mixing? There are in some cases at least four processes involved: (1) Acceleration of the particles of secondary fluid by impact of particles from the primary jet. (2) Entrainment of secondary fluid by viscous friction at the periphery of the primary jet. (3) Over-expansion of the primary jet to a pressure below that of the secondary fluid, with consequent flow of secondary fluid toward the axis of the jet. (4) Change of state, as in a steam injector, during which a large reduction in volume occurs by condensation, and the energy of latent heat is made available.

Theoretical analysis of a jet pump in which two or more of these processes are involved becomes very complicated. Hence the designs have been based largely on the results of experiments and tests. By plotting curves of pump efficiency, pressure and capacity, the effects of design changes can be studied.

AVAILABLE DATA ON JET-PUMP DESIGN

A brief discussion, with references, illustrates the possibilities of design and performance of the single-stage jet pump. Two classes of jet pumps might be recognized, *volume pumps* and *pressure pumps*. A similar classification is frequently applied to centrifugal pumps and fans. The volume pump is usually a large unit, or at least the venturi mixer is large as compared to the nozzle. The most common use of the pressure pump is as a vacuum pump, although the discharge pressure may occasionally be even higher than the jet-supply pressure, as in the locomotive injector. Most high-pressure pumps depend on the condensing of a vapor, and the venturi element is often compounded.

TABLE 2—CALCULATED AIR-STREAM PERFORMANCE
(*Assuming Uniform Velocity Across the Stream*)

OUTLET		ASPECT RATIO	ASSUMED JET ANGLE, DEG	OUTLET VELOCITY FPM	TOTAL THROW (V = 50 FPM)	TOTAL ENTRAINMENT (AT V = 50 FPM)	ENTRAINMENT PER FOOT OF STREAM, PER CENT
Area, Sq Ft	Size, In.						
0.50	8.48 × 8.48	1	19	400	14.8	7	47.2
0.50	8.48 × 8.48	1	19	800	31.7	15	47.2
0.50	8.48 × 8.48	1	19	1200	48.5	23	47.2
0.50	12 × 6	2	19	400	14.7	7	47.5
0.50	12 × 6	2	14	800	42.9	15	35.0
0.50	12 × 6	2	19	800	31.6	15	47.5
0.50	12 × 6	2	22	800	27.2	15	55.0
0.50	12 × 6	2	30	800	19.7	15	76.0
0.50	12 × 6	2	60	800	9.2	15	63.0
0.50	12 × 6	2	90	800	5.3	15	83.0
0.50	24 × 3	8	19	800	30.6	15	49.0
0.167	4.9 × 4.9	1	19	800	18.3	15	82.0
0.167	24 × 1	24	19	800	16.6	15	90.5
0.083	24 × ½	48	19	800	11.1	15	135.0

Table 1 shows typical design proportions for a number of single-stage pumps (see references). Most of these would be classified as volume pumps, but it is evident that even for similar applications there is a wide flexibility in the design. The diameter of the venturi throat is usually made 2 to 4 times the diameter of the nozzle. The venturi mixer must be accurately centered, with its throat at a distance of 4 to 20 nozzle diameters from the face of the nozzle. The venturi must be large enough so that the nozzle jet, which has a natural angle of around 20 deg, will not produce objectionable *splashing* at the inlet. The entrance to the venturi is either a rounded nozzle or a cone of about 20 deg angle followed by a short parallel section. The angle of the expanding diffuser beyond the venturi throat is usually between 4 deg and 10 deg and its length may be as little as twice its smallest diameter if necessary, although a longer diffuser is preferred for pressure recovery. Most investigators report improved performance of the pump as the diameter and length of the venturi mixer are increased, and the limitations are mainly those of space and cost. Some commercial devices use very small nozzles in multiple, or slot-type nozzles instead of round ones.

TABLE 3—TOTAL MOMENTUM OF AIR STREAMS
(*First Series*)

OUTLET SIZE	DISTANCE FROM OUTLET IN FT	FACE VELOCITY, FPM (CFM DIVIDED BY GROSS AREA)			
		400	800	1200	1600
12 in. × 6 in. Free Opening, Duct Approach	0	0.050	0.196	0.458	0.80
	3	0.051	0.200	0.428	0.78
	6	0.053	0.183	0.427	0.81
	9	...	0.184	0.412	0.71
	12	...	0.190	0.408	0.74
12 in. × 6 in. Bar Grille, Duct Approach	0	...	0.1155	...	0.462
	3	...	0.0850	...	...
	6	...	0.1018	...	0.324
	9	...	0.0935	...	0.332
	12	...	0.0995	...	0.336
12 in. × 6 in. Nozzle Grille, Duct Approach	0	...	0.0606	...	0.2430
	3	...	0.0401	...	0.1935
	6	...	0.0468	...	0.1562
	9	...	...	...	0.1500
	12	...	...	...	0.1760
12 in. × 6 in. Square-Punched Grille, Duct Approach	0	0.0254	0.1019	0.2300	0.403
	3	0.0273	0.0945	0.1960	0.416
	6	0.0277	0.1039	0.2052	0.379
	9	...	0.0982	0.2120	0.380
	12	...	0.0918	0.2235	0.429
	15	...	...	...	0.395
	18	...	...	...	0.404
20 in. × 8 in. Free Opening, Duct Approach	0	...	0.439	0.998	1.768
	3	...	0.335	0.944	...
	6	...	0.400	0.825	1.555
	9	...	0.404	...	1.600
	12	...	0.403	...	...
20 in. × 8 in. Bar Grille, Duct Approach	0	...	0.290	0.530	1.185
	3	...	0.245	...	1.058
	6	...	0.275	0.563	1.060
	9	...	0.266	...	0.915
	12	...	0.260	...	0.990
14 in. × 4 in. Free Opening, Duct Approach	0	...	...	0.327	...
	3	...	...	0.279	...
	6	...	...	0.284	...
	9	...	...	...	...
	12	...	...	0.286	...
12 in. × 12 in. Free Opening, Duct Approach	0	...	0.388	...	...
	3	...	...	...	...
	6	...	0.335	...	...
	9	...	0.321	...	...
	12	...	0.336	...	...
12 in. × 6 in. Free Opening, Plenum Approach	0	0.1969	...	0.442	0.786
	3	0.2050	...	0.520	0.903
	6	0.2305	...	0.440	0.878
	9	0.2360	...	...	0.874
	12	...	...	0.525	...

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TABLE 4—TOTAL MOMENTUM OF AIR STREAMS
(*Second Series*)

OUTLET SIZE	DISTANCE FROM OUTLET IN FT	FACE VELOCITY, FPM (CFM DIVIDED BY GROSS AREA)			
		400	800	1200	2400
8.48 in. × 8.48 in.	0	0.0750	0.278	0.681	...
Free Opening,	9	0.0764	0.276	0.539	...
Plenum Approach					
10.285 in. × 7 in.	0	0.0712	0.282	0.680	...
Free Opening,	9	0.0748	0.301	0.585	...
Plenum Approach					
12 in. × 6 in.	0	0.737	0.299	0.644	1.828
Free Opening,	9	0.0681	0.259	0.557	1.640
Plenum Approach					
14.4 in. × 5 in.	0	0.0666	0.297	0.648	...
Free Opening,	9	0.0651	0.299	0.529	...
Plenum Approach					
18 in. × 4 in.	0	0.0666	0.296	0.648	...
Free Opening,	9	0.0625	0.285	0.519	...
Plenum Approach					
24 in. × 3 in.	0	0.0695	0.290	0.648	...
Free Opening	9	0.0657	0.301	0.560	...
Plenum Approach					
24 in. × 2 in.	0	...	0.187	0.431	...
Free Opening,	9	...	0.215	0.406	...
Plenum Approach					
24 in. × 1 in.	0	...	0.1040	0.192	0.823
Free Opening,	6	...	0.1005	0.192	0.668
Plenum Approach	9	...	0.1155	0.202	0.620
24 in. × ½ in.	0	...	0.480	0.105	0.416
Free Opening,	6	...	0.570	0.111	0.351
Plenum Approach	9	...	0.456	0.107	0.344
4.9 in. × 4.9 in.	0	...	0.1153	0.212	0.823
Free Opening,	6	...	0.1030	0.190	0.748
Plenum Approach	9	...	0.1160	0.204	0.642

When the pressure difference across the venturi is increased, either by increasing the entrainment vacuum or by increasing the exit resistance, the pump becomes more expensive to operate because a much higher jet pressure is required. There is no easy way to calculate the jet pressure required for a given case. Trinks [1] has derived an equation for computing the nozzle jet velocity required to produce a specified pressure and velocity of the mix-

[1] Industrial Furnaces, by W. Trinks (Vol. 2, p. 54.)

TABLE 5—ENTRAINMENT RATIOS FOR FREE AIR STREAMS

OUTLET SIZE	DISTANCE FROM OUTLET IN FT	FACE VELOCITY, FPM (CFM DIVIDED BY GROSS AREA)			
		400	800	1200	1600
12 in. × 6 in. Free Opening, Duct Approach	3	1.77	1.62	1.69	1.67
	6	3.10	2.98	3.17	2.83
	9	...	4.19	4.51	4.17
	12	...	5.58	6.45	5.44
12 in. × 6 in. Bar Grille, Duct Approach	3	...	1.45	...	...
	6	...	2.76	...	2.40
	9	...	4.10	...	3.82
	12	...	5.40	...	5.25
12 in. × 6 in. Nozzle Grille, Duct Approach	1.5	...	1.17	...	1.38
	3	...	1.985	...	1.985
	6	...	3.645	...	3.42
	9	...	...	...	4.60
	12	...	...	...	6.81
12 in. × 6 in. Square Punched Grille, Duct Approach	1.5	1.13	1.03	1.03	...
	3	1.92	1.96	1.68	1.94
	6	3.56	3.56	3.52	3.58
	9	...	5.36	5.08	5.13
	12	...	6.46	7.39	7.56
	15	...	...	...	8.84
	18	...	...	...	11.2
20 in. × 8 in. Free Opening, Duct Approach	3	...	1.06	1.16	...
	6	...	2.50	1.85	1.98
	9	...	3.04	...	2.94
	12	...	3.91	...	...
20 in. × 8 in. Bar Grille, Duct Approach	3	...	1.28	...	0.93
	6	...	2.18	1.78	1.91
	9	...	2.91	...	2.68
	12	...	3.82	...	3.80
14 in. × 4 in. Free Opening, Duct Approach	3	...	...	1.58	...
	6	...	...	3.13	...
	9	...	...	...	...
	12	...	...	5.74	...
12 in. × 12 in. Free Opening, Duct Approach	3	...	...	...	...
	6	...	2.00	...	...
	9	...	2.66	...	...
	12	...	3.68	...	...
12 in. × 6 in. Free Opening, Plenum Approach	3	1.88	...	1.78	1.90
	6	3.99	...	3.54	3.46
	9	5.42	...	...	5.40
	12	...	...	6.82	...

ture at the diffuser outlet. His analysis was based on conservation of momentum and conservation of energy, with an assumed efficiency of pressure recovery in the diffuser of 50 per cent. The application was that of a gas burner mounted in the wall of a furnace, with a pressure of 0.025 in. of water in

the furnace. Howe and Johnson[2] have used a similar analysis to derive a family of curves covering the range of nozzle supply pressures from 3 lb to 60 lb per sq in. gage, and for resistances to 2 in. of water. These curves were checked experimentally with reasonable success.

The pressure-volume relationships for a jet pump may be likened to the family of characteristic curves for a centrifugal fan. In both cases the shape of these curves, between wide-open and blocked-tight (S.N.D.) conditions,

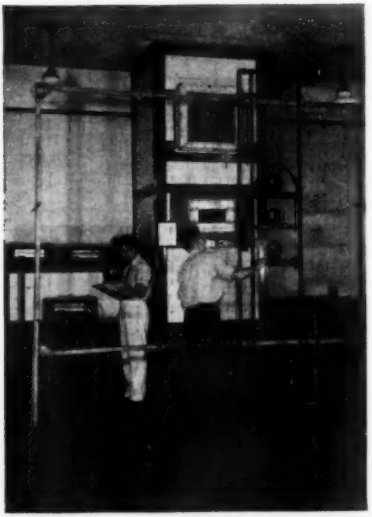

FIG. 2. VIEW OF TRAVERSE-GRID FOR OBSERVING AIR MOVEMENT

are affected by the design proportions. One of the authors has shown, by constructing and testing a series of jet pumps that by changing the ratios D_2/D_1 and M/D_1 (Table 1), the curve of nozzle pressure against vacuum for a high-pressure air ejector may be widely varied to suit the application.[3] But several investigators[4,5,6] have reported that with the lower jet pressures (below 5 lb gage) and the usual design proportions, the entrainment ratio is only slightly affected by changing the jet pressure.

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[2] Atmospheric Type Burners Used with Natural Gas, by Howe and Johnson. (*ASME Transactions*, November, 1939, p. 673.

[3] Design of Air-Atomizing Oil Burners, by G. L. Tuve. (*M. E. Thesis*, University of Minnesota, 1921.)

[4] A Study of the Locomotive Front-End, by E. G. Young. (*Bulletin University of Illinois*, Engineering Experiment Station, No. 256. 1933, 180 pp.)

[5] Efficiencies and Entrainment Ratios of Air Jet Pumps, by G. P. Miller. (*Master's Thesis*, Case School of Applied Science, 1940, 77 pp.)

[6] The Performance of Jet Pumps, by Herrick and Horsburgh. (*Thesis*, Case School of Applied Science, 1940.)

For the design of fume exhausters, curves are given in Fan Engineering [7] which show the effect of changes in D_2/D_1 upon the entrainment ratio, the pressure and the efficiency. The accompanying text states that the pressure-capacity curve plotted over the range from static-no-delivery to full-open is substantially a straight line.

From the foregoing discussion of the single-stage jet pump with venturi, some conclusions may be drawn for application to the devices used for air

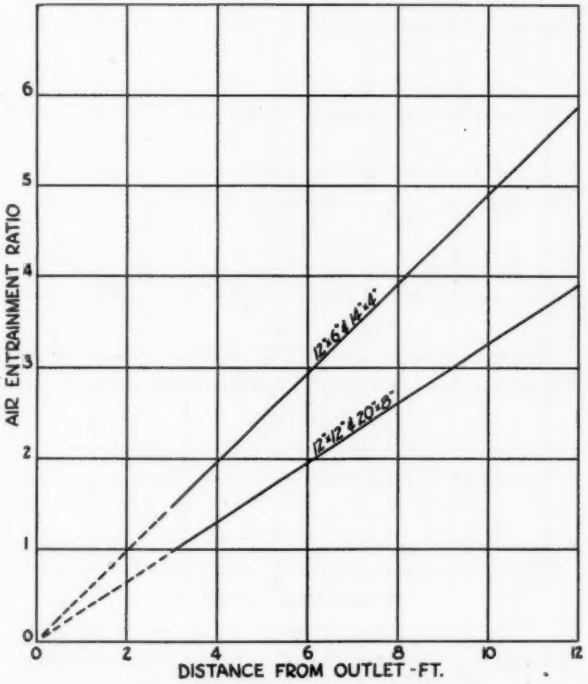

FIG. 3. EFFECT OF OUTLET AREA ON AIR ENTRAINMENT RATIO; FREE OPENING AT END OF DUCT; FACE VELOCITIES 400 TO 1600 FPM (FOR TEST POINTS SEE TABLE 5)

mixing in rooms. If such a room air diffuser utilizes a venturi mixer, the entrainment within this mixer will probably be less than twice the primary air volume, due to the prohibitive size of the venturi for large entrainment. Entrained volume will be nearly proportional to the velocity of the primary jet (entrainment ratio constant), but the noise factor will limit the jet velocity which may be used. The entrainment might be increased by using small nozzles or narrow slots.

[7] Fan Engineering, Buffalo Forge Co., 4th edition, p. 229.

JETS IN FREE-OPEN SPACE

When a jet discharges into free-open space without a venturi mixer, the case is less complicated and is more easily attacked by theoretical analysis. Many such analyses have been reported, based on theories of diffusion or of turbulence, and involving various assumptions or approximations. Among the common simplifying assumptions are: (1) That the static pressure throughout the open jet is constant, *i.e.*, no vacuum exists. (2) That the viscous

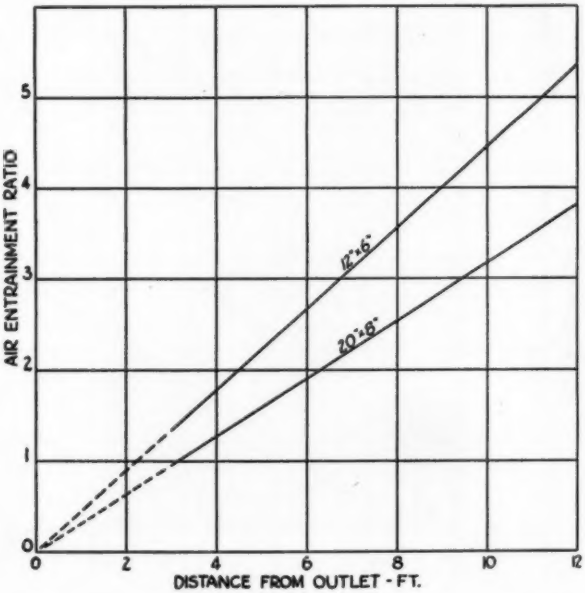

FIG. 4. EFFECT OF GRILLE SIZE ON AIR ENTRAINMENT RATIO; BAR GRILLES, OTHERWISE IDENTICAL; FACE VELOCITIES 800 TO 1600 FPM (FOR TEST POINTS SEE TABLE 5)

forces are small as compared with the turbulent or inertia forces. (3) That the primary and secondary fluids are of equal density. (4) That the jet is being discharged from a point source. (5) That the jet is entirely symmetrical. (6) That the velocity across the jet is uniform. These assumptions are all closely approximated in such a practical case as that of a small gas burner mixer, or in any low-pressure device where gas or air discharges from a small, circular, rounded-entrance nozzle into the atmosphere.

For the specific application of air discharged from an outlet opening or grille into free-open space in a room, the assumptions of a point source, jet symmetry and uniform velocity cannot be used. There is little error in assuming the pressure constant, *i.e.*, a negligible vacuum within the jet. More-

over, the viscous drag is small on account of the low viscosity of air, and the densities are practically equal. Re-examining the four mixing processes already enumerated, it is found that only one remains, that of impact of the primary stream upon the free air in the room. The theory of conservation of momentum is then applicable. The collisions of particles may be studied in accordance with the statistical theories of turbulence,[8] but since experimental veri-

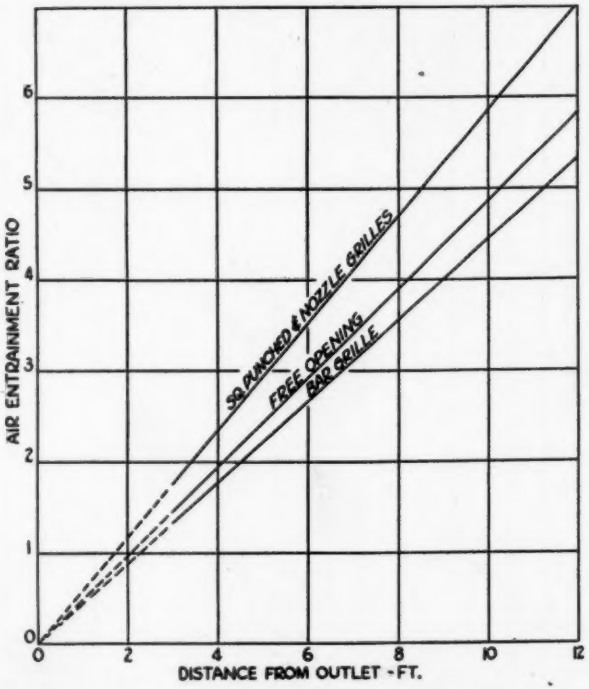

FIG. 5. EFFECT OF TYPE OF OUTLET ON AIR ENTRAINMENT RATIO; 12 IN. X 6 IN. OUTLETS AT END OF DUCT; FACE VELOCITIES 400 TO 2000 FPM (FOR TEST POINTS SEE TABLE 5)

fication and evaluation of constants are necessary in any case, it may be more direct to use a method of experimental integration instead of carrying out an analysis of the mechanisms of turbulent diffusion.

The mixing-space available with a free-open jet is greatly increased over that of a jet discharging to a venturi; hence a greater entrainment ratio might be expected. But there are few published data giving quantities of air actually entrained by open jets. That the entraining or mixing action is

[8] Investigations of Turbulent Mixing Regions Formed by Jets, by A. M. Kuethe. (*Journal of Applied Mechanics, ASME*, 1935, p. A-85.)

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continuous along the path of the jet has been shown by tests of propeller fans. In fact, this principle is turned to commercial advantage when fans are rated by the *NEMA* Code.[9] A series of such tests made by the authors on six 4-bladed propeller fans from 24 in. to 50 in. wheel diameter showed an average rating of about 125 per cent of primary air volume at 12 in. in

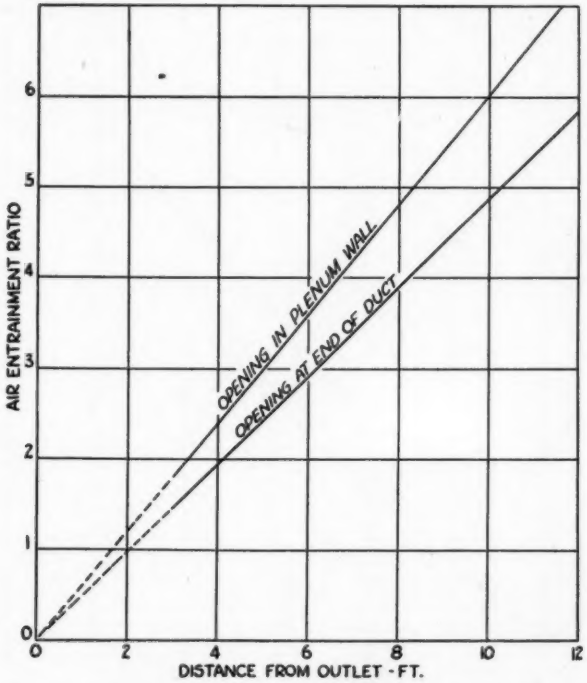

FIG. 6. EFFECT OF TYPE OF APPROACH ON AIR ENTRAINMENT RATIO; 12 IN. X 6 IN. FREE OPENINGS; FACE VELOCITIES 400 TO 1600 FPM (FOR TEST POINTS SEE TABLE 5)

front of the fan, and about 155 per cent of primary air volume at 48 in. in front of the fan. One of the methods for increasing the entrainment of air by a simple jet is to use a slot-type outlet. The use of narrow slots for room air distribution has been discussed by Mackey,[10] Stevenson [11] and others.

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[9] *NEMA* Fan Standards. Publication No. 39-56, *National Electrical Manufacturers' Association*, New York, N. Y., 1939.

[10] Air Conditioning Principles, by C. O. Mackey. (International Textbook Co., 1941, p. 184.)

[11] Air Discharge from Narrow Slots, by F. F. Stevenson. (*Heating, Piping and Air Conditioning*. May and June, 1941.)

One of the important quantities to be determined for a room-supply air stream is the total blow or throw of the primary jet. As a result of both analysis and experiment [10,12,13] Mackey and others have stated the total throw in terms of the area of the outlet and the primary air volume (or the face velocity):

$$T = \frac{KQ}{\sqrt{AB}} = KV\sqrt{AB} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where T is the total throw, Q the primary air volume, V the face velocity, and A and B the linear dimensions (assuming a rectangular outlet). Additional experimental results in support of this equation would be desirable.

THE MOMENTUM THEORY

The mixing action of an open jet is due almost entirely to the impact process, as has already been indicated, and by simple application of the theory of conservation of momentum, many of the performance laws for a free-open stream may be developed. The assumptions are those already enumerated: Uniform density, pressure, temperature and velocity, and negligible viscosity. The assumption of a point source is not necessary, and jet symmetry is assumed only with respect to the angle of spread, not to the opening itself. Among these assumptions, the most serious violation of practical conditions lies in the statement that uniform velocity exists across any section perpendicular to the stream, *i.e.*, at the outlet or parallel to it.

Constant momentum of the stream ($MV = \text{constant}$) means also that the total quantity of air in the moving stream is inversely proportional to the stream velocity. This is demonstrated as follows:

$$\begin{aligned} M_1V_1 &= M_2V_2 \\ wQ_1V_1 &= wQ_2V_2 \\ \frac{Q_2}{Q_1} &= \frac{V_1}{V_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3) \end{aligned}$$

where Q is the total quantity or volume of air moving, at sections 1 and 2, V is its uniform velocity at that section, and w is its constant density. From this equation we can obtain the air entrainment at any plane where the velocity is known. If we arbitrarily define the end of the *throw* as that section where the average velocity has reached, say, 50 fpm, the total air in motion at the end of the throw for an outlet velocity of 1000 fpm, is 20 times the primary air volume. For an outlet velocity of 2000 fpm the total air mixture at the end of the throw would be 40 times the primary volume.

This does *not* mean that at a given plane, say, 10 ft from the outlet, the ratio of entrained room air to primary air increases as the face velocity increases. In fact, the entrainment ratio at a fixed distance from the outlet

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[12] Air Paths from Grilles, by Greenlaw and Hart. (*Heating, Piping and Air Conditioning*, June and July, 1938.)

[13] Measuring Air Distribution and Grille Performance in Air Conditioning, by G. L. Tuve. (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, November, 1937.)

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should be independent of the outlet velocity, as shown by the following derivation:

Since

$$Q = AV, \text{ or } V = \frac{Q}{A},$$

Equation (3) becomes:

$$\frac{Q_2}{Q_1} = \frac{Q_1 A_2}{A_1 Q_2}$$

or

$$\frac{Q_2}{Q_1} = \sqrt{\frac{A_2}{A_1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where A is the area of the stream at the section under consideration and Q is the total volume of air moving. But the ratio A_2/A_1 is dependent on the stream angle only, and it is assumed that this angle does not change as the jet velocity is varied. Hence the entrainment ratio at a given distance from the outlet should be independent of the velocity at the outlet face.

Mackey [14] has made a number of analyses based on the momentum theory, with the assumption of uniform velocity across the stream. He has derived equations for the throw and the entrainment of a free stream. His examples deal with both straight-flow and wide-angle streams from rectangular outlets of various aspect ratios. Some numerical results based on these ideal equations are given in Table 2.

Analytical predictions of jet performance such as those given in Table 2, whether based on the momentum theory or otherwise, are all dependent on many simplifying assumptions. Hence they lead to several questions which can be answered only by experiment:

1. Does the total momentum of the stream actually remain constant?
2. Is the ratio of entrained room air to primary air at a given distance from the outlet actually independent of face velocity?
3. Is the throw actually proportional to the face velocity?
4. What are the effects of variations of velocity across the stream, and is there any practical way of taking these effects into account?
5. What are the effects of the type of approach to an outlet?
6. What is the actual angle of spread of the jet from a rectangular outlet, and is this angle affected by the velocity or quantity of air discharged from a given outlet?
7. Can these analyses be applied to wide-spread grilles, ceiling outlets, and outlets of any shape, size or aspect-ratio?

In an effort to answer some of these questions, the following experimental program was undertaken.

EXPERIMENTAL SET-UP AND PROCEDURE

The first requirement for checking the momentum theory was to devise some means for taking into account the variations in velocity across an actual air stream. The analytical predictions of Table 2 were computed on the assump-

[14] The Rationale of Air Distribution and Grille Performance, by C. O. Mackey. (*Refrigerating Engineering*, June, 1938.)

tion of *uniform* velocity at all points in a plane at right angles to the stream. But when measurements are made across an actual stream, a wide variation in velocities is found. By dividing the stream into small areas and taking velocity measurements within each small area it is possible to remove this chief violation of actual conditions. In this way a much more accurate application of the momentum theory can be made.

Fig. 2 shows the apparatus used for such a stream analysis on the basis of *differential areas*. By means of a grid of 6-in. squares formed by threads stretched across a large frame, the stream was explored with a velometer.

8.48"×8.48" FREE OPENING VEL.×AREA × 1200
TEST NO. 20 DATE 8-13-41 DISTANCE - 9 FT.
OBSERVERS: GBP&HWE

			0	0	0	0	0	0			
		0	90	90	130	80	50	20	0		
	0	10	170	170	190	200	210	60	0	0	
	0	150	280	300	300	290	300	160	40	50	0
	0	160	210	290	320	300	290	240	110	60	0
0	10	120	180	310	470	440	300	300	130	50	0
	0	110	150	300	400	360	290	270	160	70	0
	0	50	100	300	300	300	300	210	120	20	0
		0	50	200	150	180	200	130	60	0	
			0	10	50	90	70	20	10	0	
				0	0	10	0	10	0		
					0	0	0	0			

FIG. 7. TYPICAL DATA SHEET SHOWING MAXIMUM VELOCITY OF AIR AS MEASURED BY TRAVERSE-GRID METHOD

A velocity reading was taken in the center of each 6-in. square where the crepe-paper ribbons indicated air movement (see Fig. 2). Since the mass of the air flowing in any small square is wAV, the momentum is:

$$MV = wAV^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

That is, the momentum is equal to the product of the air density, the area, and the velocity squared. These individual momenta were computed for each small square, and the sum of all of them represented an integrated momentum or total momentum for the entire stream. This amounts to a step by step integration based on area increments of 0.25 sq ft.

Velocity measurements were taken over cross-sections of the stream spaced at intervals of 3 ft up to 18 ft from the outlet. The investigation covered several sizes of openings, several types of grilles and types of approach.

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Tests were made at various face velocities from 400 to 2400 fpm. All tests were made in a closed room 48 ft x 21 ft x 13 ft high, the air outlet discharging horizontally from one end of the room, about midway between floor and ceiling. The primary air was measured by the same calibrated orifices used in previous investigations, and the air supply system was also the same.[15,16] The test ducts and plenum chamber were frequently checked for leakage (which was always negligible). In addition to the tests on free openings and grilles in the sizes 12 in. x 6 in., 14 in. x 4 in., and 20 in. x 8 in.,

24"-1" FREE OPENING VEL. = CFM/AREA = 1200
TEST NO. 26 DATE 8-16-41 DISTANCE - 9 FT.
OBSERVERS: G.B.P. & H.W.E.

			0	0	0	0	0	0			
		0	30	40	20	10	0	50	0		
		0	60	90	100	90	50	40	0		
		0	100	150	170	160	70	100	30	0	
	0	10	110	150	210	180	100	110	50	0	
	0	20	130	180	250	250	150	90	50	0	
0	10	10	140	170	270	250	170	80	40	0	
	0	0	170	170	300	290	180	50	20	0	
	0	20	140	140	250	240	160	70	10	0	
	0	50	90	120	170	210	120	50	40	0	
		0	40	70	120	120	120	0	0		
			0	20	50	80	60	0			
				0	0	0	0				

FIG. 8. TYPICAL DATA SHEET SHOWING MAXIMUM VELOCITY OF AIR AS MEASUREI BY TRAVERSE-GRID METHOD

tests were also made on square orifices of 4 in. to 12-in. sizes, and on slot-type outlets without grilles, down to one-half inch slot width.

ACKNOWLEDGMENT

The actual tests extended over about six months of steady work, the observers working two at a time. The authors wish to extend special thanks to the following men who were responsible for most of the thousands of observations and calculations here reported: Messrs. R. E. Bolz, Yale University, W. C. Roberts, Lehigh University, D. W. Steel, H. W. Engelman and F. E. Marble, Case School of Applied Science.

[15] ASHVE RESEARCH REPORT No. 1140—The Use of Air Velocity Meters, by G. L. Tuve, D. K. Wright, Jr., and L. J. Seigel. (ASHVE TRANSACTIONS, Vol. 45, 1939.) (See Fig. 4.)
[16] ASHVE RESEARCH REPORT No. 1163—Air Flow Measurements at Intake and Discharge Openings and Grilles, by G. L. Tuve and D. K. Wright, Jr. (ASHVE TRANSACTIONS, Vol. 46, 1940.)

Tests were frequently repeated by different observers, to check the accuracy of results. It was found that while the velocities read in each individual square at different times or by different observers did not agree perfectly, the summation of velocities and summation of momenta over the entire air stream showed very good agreement. As a further precaution several different velometers were used, each velometer being calibrated during the tests. When these calibration corrections were made, the results with different velometers showed satisfactory agreement.

DEFINITIONS AND CALCULATIONS

The test results have been computed in terms of total momentum, jet angle or spread, total throw and total entrainment. *Total volume* at any cross-section of the stream was obtained as the summation of the AV products for all the 6-in. squares and *total momentum* was the summation of the products wAV^2. See Equation (5).

Entrainment ratio is defined as the ratio of the volume of room air entrained to the volume of primary air leaving the outlet. *Total entrainment* is the entrainment ratio measured at the end of the total throw.

Total throw (or blow) is the distance measured perpendicular to the outlet face, at which the velocity has reached a predetermined low value.

Jet angle or spread is the total angle in degrees between the two straight lines on opposite sides of the stream which are drawn from the edge of the outlet and tangent to the edge of the stream. (The spread becomes somewhat less than this as the end of the throw is approached.)

EXPERIMENTAL DATA AND RESULTS

A tabulation of the total momentum obtained at each test location is summarized in Tables 3 and 4. From these data, the experimental results are seen to substantiate the principle of constant momentum. Since this theory has been substantiated by others also, it is concluded that the accuracy of the method for taking the velocity pattern into account is satisfactory. In general, the mean deviation of the individual momenta from the average of all momenta for the particular test is less than 5 per cent. In view of the unsteady and surging nature of the air stream in a room, this is considered very close agreement. (*Note:* The momentum at zero distance cannot be obtained by the same method as the others; hence it may differ somewhat.)

In terms of the entrainment ratio, or the ratio of entrained room air to primary air, Table 5 shows the results of the same tests that were covered by Table 3. From Table 5 it may be seen that for practical purposes the entrainment ratio may be considered independent of velocity. But the entrainment ratio does vary with the size and shape of the discharge opening, and with the type of grille and the type of approach. A lower entrainment ratio with a larger opening is definitely indicated by the test results of Figs. 3 and 4, although the range of sizes is not great enough to give a definite relation. From these data, the larger opening would also be expected to give a longer total throw with the same outlet velocity, which confirms the results of other investigations and the theoretical calculations of Table 2.

Fig. 5 presents information upon the effect of type of grille upon entrainment ratio by comparing three different styles of grilles and also a free open-

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TABLE 6—ENTRAINMENT RATIOS FOR FREE OPENINGS
(*Plenum Approach*)

OUTLET SIZE (AREA = 0.5 SQ FT)	ASPECT RATIO	ENTRAINMENT RATIO AT 9 FT FACE VELOCITY, FPM (CFM DIVIDED BY GROSS AREA)		
		400	800	1200
8.48 in. × 8.48 in.	1.0	4.14	4.36	4.52
10.285 in. × 7 in.	1.47	4.42	4.93	4.70
12 in. × 6 in.	2.0	4.10	4.25	4.45
14.4 in. × 5 in.	2.9	4.01	4.71	4.27
18 in. × 4 in.	4.5	4.66	4.30	4.20
24 in. × 3 in.	8.0	4.27	4.81	4.77

TABLE 7—ENTRAINMENT RATIOS FOR FREE-OPEN SLOTS
(*Plenum Approach*)

SIZE	ASPECT RATIO	6 FT FROM OUTLET			9 FT FROM OUTLET		
		At 200 cfm Volume	At 800 fpm Velocity	At 1200 fpm Velocity	At 200 cfm Volume	At 800 fpm Velocity	At 1200 fpm Velocity
24 in. × 3 in.	8	...	...	...	4.27	4.81	4.77
24 in. × 2 in.	12	...	...	...	6.23	6.61	6.27
24 in. × 1 in.	24	6.67	6.81	6.67	8.81	8.12	8.81
24 in. × ½ in.	48	9.20	9.07	10.10	11.98	11.43	12.15

TABLE 8—ENTRAINMENT RATIOS FOR FREE OPENINGS
(*Plenum Approach*)

SIZE (AREA 24 SQ IN.)	ASPECT RATIO	6 FT FROM OUTLET			9 FT FROM OUTLET		
		800 fpm	1200 fpm	2400 fpm	800 fpm	1200 fpm	2400 fpm
4.9 in. × 4.9 in.	1	5.01	4.89	4.72	7.47	7.13	6.88
24 in. × 1 in.	24	6.81	6.67	6.00	8.12	8.81	8.38

TABLE 9—DISTANCES FROM OUTLET IN FEET WHERE MAXIMUM VELOCITY WAS 100 FPM

OUTLET SIZE	AREA SQ FT	ASPECT RATIO	VELOCITY = CFM DIVIDED BY GROSS AREA				
			400 fpm	600 fpm	800 fpm	1200 fpm	2400 fpm
8.48 in. × 8.48 in.	0.5	1.0	18	...	32	...	...
10.285 in. × 7 in.	0.5	1.47	18	...	36	...	...
12 in. × 6 in.	0.5	2.0	18	...	36	...	...
14.4 in. × 5 in.	0.5	2.9	18	...	35	...	...
18 in. × 4 in.	0.5	4.5	17	...	32	...	...
24 in. × 3 in.	0.5	8.0	16	...	32	...	...
4.9 in. × 4.9 in.	0.167	1	...	17	24	35	...
24 in. × 1 in.	0.167	24	...	...	18	27	...
24 in. × ½ in.	0.083	48	...	...	12	18	36

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ing, all of 12 in. x 6 in. size. Fig. 6 shows the difference between a straight-duct approach and a plenum approach. This figure may also be taken as a comparison between zero velocity of approach to the outlet, as when the grille is mounted in the wall of a plenum chamber, and maximum approach velocity, as when the grille is mounted at the end of a long duct of the same nominal size. The conclusion might be drawn from Figs. 5 and 6 that streams from square-edged orifice type openings tend to give slightly higher entrainment ratios than streams which are guided by ducts and bars, but the difference is not over 20 per cent.

The effect of the shape of the outlet, expressed as aspect-ratio or as slot width, is indicated by Tables 6 to 8. Table 6 reports tests of six free openings of equal area but of various shapes. These tests show that for ordinary rectangular openings the aspect ratio or shape factor has little effect upon the air entrainment. This agrees with the computed data in Table 2. Table 7 carries the study farther, and gives results on free-open slots from 3 in. wide down to ½ in. wide. Here it is demonstrated that by reducing the slot width to less than 2 in. the entrainment may be markedly increased. Table 8 is a comparison of a 1-in. slot and a square outlet, both of the same total area. The effect of changing the aspect ratio is not as great as might be expected.

Limitations of the testing space available made it difficult to measure the total throw. Table 9 gives the results of a few tests in which the face velocity was varied and the total throw was measured. In these tests the end of the throw was defined as the plane at which the maximum velocity measured by the velometer was 100 fpm.

This relation between average velocity and maximum velocity is of much practical importance, especially when measuring total throw or in specifying conditions which shall be met in order to eliminate *drafts*. The results of extensive studies show that the maximum velocity is usually from 2.5 to 3.5 times the average velocity across the section as measured by the traverse-grid method.

Typical data sheets from which these and the other results were obtained are shown in Figs. 7 and 8. These sheets give the uncorrected velometer readings as observed at the grid, and the outlet size is superimposed upon each traverse area, in proper position and to scale.

Jet angles were measured in all tests, and Table 10 gives a summary of some of the results. Angles were determined experimentally from layouts to scale, using data sheets, similar to Figs. 7 and 8 for distances of 3 to 12 ft from the outlet. The *average* jet angle, which is taken as the arithmetical mean between the horizontal and the vertical angle of spread, tends to increase slightly with increasing outlet velocity. For a slot-type outlet the horizontal spread is not the same as the vertical spread, and the stream tends to become circular. A few tests were made on directional-flow grilles with wide-spreading stream, and these indicated that very much greater air entrainment may be obtained with this type of outlet, as predicted by the results in Table 2.

Such items as directional flow grilles and heating and cooling of the air stream, should be covered by further investigation. These studies are to be continued under the direction of the Technical Advisory Committee on Air Distribution and Air Friction of the ASHVE, as already authorized by the

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Research Committee of the Society. Critical comment and suggestions for this work will be gladly received.

CONCLUSIONS FROM TESTS

The following conclusions may be drawn as a summary of the experimental results here presented. It will be noted that the tests were limited to air outlets less than 160 sq in. in area and outlet velocities up to 2400 fpm, but that a variety of types and sizes were tested.

TABLE 10—JET ANGLES IN DEGREES

FREE OPENINGS, PLENUM APPROACH	FACE VELOCITY, FPM (CFM DIVIDED BY GROSS AREA)								
	400			800			1200		
	Hor	Vert	Av	Hor	Vert	Av	Hor	Vert	Av
8.48 in. × 8.48 in.	21	20	20.5	23	25	24	24	28	26
10.285 in. × 7 in..	20	20	20	22	26	24	23	28	25.5
12 in. × 6 in.....	17	23	20	18	26	22	20	28	24
14.4 in. × 5 in....	16	24	20	17	29	23	21	30	24.5
18 in. × 4 in.....	14	26	20	14	28	21	16	31	23.5
24 in. × 3 in.....	10	31	20	14	30	22	15	34	24.5
4.9 in. × 4.9 in...	20	22	21	20	22	21	22	21	21.5
24 in. × 1 in.....	10	26	18	10	28	19	14	32	23
24 in. × ½ in....	10	30	20	11	31	21	12	32	22

1. The total momentum of a free air stream remains practically constant throughout its measurable length.
2. Entrainment of room air is continuous throughout the length of an air stream, and amounts to 20 to 70 per cent of the primary air volume per foot of travel or throw. This percentage entrainment may be increased greatly by using slot-type outlets or wide-angle spreading grilles.
3. The ratio of entrained air to primary air *at a given distance* from the outlet face varies as follows:
   (a) Entrainment ratio decreases as the area of the outlet is increased.
   (b) Entrainment ratio is practically independent of the outlet face velocity.
   (c) Entrainment ratio is practically the same for all square and rectangular outlets of the same area, except the very narrow slots.
4. The total throw or blow of an air stream is directly proportional to the outlet velocity.
5. The air stream from a narrow-slot outlet becomes almost circular toward the end of its throw.
6. The maximum velocity at any plane at right angles to the stream is usually about three times the average velocity in that plane.
7. The *average* jet angle for the stream from a rectangular outlet of any shape, without spreading vanes, is about 19 deg, ± 5 deg depending on type of approach, type of outlet and velocity.
8. The air entrainment by a venturi-mixer unit is small compared with the natural entrainment by a free air stream.
9. For increasing the air mixing and reducing the throw, the advantages of the narrow slot or of the narrow-ring ceiling outlet are clearly indicated, as are those of the wide-spreading grille.

DISCUSSION

W. E. ZIEBER, York, Pa. (WRITTEN): The manner in which the entrainment ratios and the total throw experimental figures conform with the calculated figures in Table 2 is very interesting. The writer believes that this paper would be more complete if the experimental data are compared directly with calculated results from the formula or if the factors were put in the theoretical formula that would make them line up with the experimental work, especially in the reduced slot widths below 3 in. The formula could be rearranged or factors applied to cover the effect when narrow slots are used that would be of considerable engineering value. It appears from the information available that there might be sufficient data to accomplish this result without going into any more experimental work.

Table 2 lists some assumed jet angles up to 90 deg and shows the effect upon total throw. Some experimental work to confirm this would be very interesting and useful to the engineering world.

Velocities at the outlet up to 2400 fpm have been used in this investigation. We would like to know whether the noise level is sufficient to eliminate these velocities in practice or whether certain openings could be used with these velocities without worrying about noise effect.

Has there been any information obtained in this work that would show how far apart two narrow slots could be placed without affecting entrainment capacity of each?

C. M. ASHLEY, Syracuse, N. Y. (WRITTEN): It seems unfortunate that some of the openings were such as to give an orifice effect so that the opening does not represent the true size of the stream. However, if the magnitude of this effect is known it should be possible to reinterpret the data on a satisfactory basis.

A second point which I cannot understand is the data on the momentum of air streams with different types of grilles. You would expect that for a given air quantity through a given face area the type of grille would have no appreciable effect upon the momentum, yet this does not appear to be the case from the data.

The third point concerns the data shown in Table 9 which seem to indicate:

(a) that the throw of a square duct is less than the throw of a rectangular duct with the same area. I do not believe that this is actually the case and wonder what comments the authors have on this particular point.

(b) Another point from the same table is that the ratio of throw between the small square opening and the large square opening does not seem to be proportional to the dimensions as should be expected and as our data all indicate it to be. If, however, the throw for the large square outlet were increased to 36 ft at 800 fpm and if the throw for the small square outlet were decreased to between 22½ and 23 ft for 800 fpm, consistent with the results for the same outlet at 600 and 1200 feet and if instead of the physical size of the outlet the throw was made proportional to the square root of the momenta then the result would check. I think this is a point of some importance because it is the only quantitative data from which the effect of size can be inferred.

Perhaps the most significant comment which I might make is that the induction ratio shown in this paper does not correspond to that obtained in practical experience. The reason for this is that in an actual application there is a considerable back flow of air counter to the air stream. This back flow results in a resistance to the forward motion of the air stream, which materially slows down the forward motion close to its terminus and, at the same time, peels off the outer layers of the stream so that they can no longer be recognized as a coherent part of it.

As a result of this effect the induction ratio is limited according to the approximate relationship, ratio total volume to primary volume of the stream equals initial velocity divided by 200. Thus, if you had an initial velocity of 1000 fpm, the induction ratio would be 5 to 1.

The figures which I have given are based upon a back flow of approximately 50 fpm. Obviously with higher return velocity through the room, the induction will be still less,

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whereas with lower velocity the induction will be greater. Incidentally in a confined room it is quite easy to measure the maximum rate of induction since this can be done by measuring the return air velocity after deducting the cross section of the stream itself from the room sectional area.

It is to be hoped that this splendid work will be followed up by a study of the effect of temperature on both horizontal and vertical air streams and this work should include not only an experimental program but also a correlation with the theory.

C. H. COOGAN † and JOHN A. GOFF, Philadelphia, Pa. (WRITTEN): The authors present experimental data which should interest everyone concerned with the entrainment problem. The theory of the turbulent mixing of jets was first developed by Tollmein [17] who applied Prandtl's [18] conception of apparent shear stress and mixing length to the problem. Tollmein examined theoretically three cases: (*a*) the plane parallel jet with mixing on one boundary only; (*b*) the plane jet; and (*c*) the rotationally symmetrical jet. In each case the jet was assumed to issue into still air having the same pressure and density, which were taken to be constant throughout the mixing region. By making certain assumptions regarding the mixing length and the similarity of the mixing process, then applying the fundamental equations of fluid mechanics with appropriate boundary conditions, Tollmein was able to predict the velocity profiles for each case, except for an empirical constant which can be determined by experiment. Tollmein, by comparison with experimental data, calculated the empirical constant for cases (*a*) and (*c*). Kuethe [19] extended Tollmein's theory to the case where the driven fluid has an initial velocity parallel to the driving fluid. The writers [20] extended the Tollmein theory to the case where the jet issues into still air and the driving and driven fluids have widely different densities.

A great many air outlets encountered in the heating, ventilating and air conditioning field can certainly be idealized to one of the cases previously mentioned, for certainly the conditions of essentially constant pressure and uniform density are customarily those encountered in the field. The empirical constant must, of course, be determined for each similar case, but this can possibly be determined from the complete experimental data of the authors.

Förthmann [21] determined experimentally the velocity profiles for air discharging from a slot 1.18 in. x 25.6 in. with a maximum discharge velocity of 113.3 fps. The results of his experiments should check closely with the data from the authors' 1 in. x 24 in. slot, tested at 40 fps maximum velocity, when plotted on a comparable basis. Förthmann also examined experimentally the partially open jet. Förthmann used a unique plot for the velocity profiles which eliminated the empirical constant. In each of the two cases which Förthmann studied the velocity profiles at various distances downstream from the slot were similar and the experimental data, except for small systematic discrepancies, check the Tollmein theory predictions.

The Tollmein theory predicts that in the open slot the maximum velocity, which occurs along the center line of the jet, varies inversely with the square root of the distance from the slot. The Förthmann data show that after certain adjustments of the distance, due to the finite slot width, that the prediction given is essenially correct. The non-dimensional method of plotting which eliminates the empirical

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[17] Berechnung turbulenter Ausbreitungsvorgänge, by W. Tollmein. (*Zeitschrift für angewandte Mathematik und Mechanik*, Band 6, Heft 6, Dec. 1926.)

[18] Bericht über Untersuchungen zur ausgebildeten Turbulenz, by L. Prandtl. (*Zeitschrift für angewandte Mathematik und Mechanik*, Band 5, 1925).

[19] Investigations of the Turbulent Mixing Regions Formed by Jets, by A. M. Kuethe. (*Transactions ASME*, vol. 57, 1935).

[20] Some Two-Dimensional Aspects of the Ejector Problem, by J. A. Goff and C. H. Coogan. (*ASME Journal of Applied Mechanics*, Vol. 9, No. 4, 1942.)

[21] Über turbulente Strahlausbreitung, by E. Förthmann Ingenieur-Archiv. Vol. 5, no. 1, 1934.

constant is accomplished by plotting as ordinate; the ratio of the x component of the velocity divided by the maximum velocity, against, as abscissa; the ratio of the y distance from the maximum velocity to the point where the x component of the velocity which is used as numerator for the ordinate value occurs, divided by the y distance from the maximum velocity, to the point where the x component of the velocity is one-half of the maximum velocity, all quantities being taken at the same x

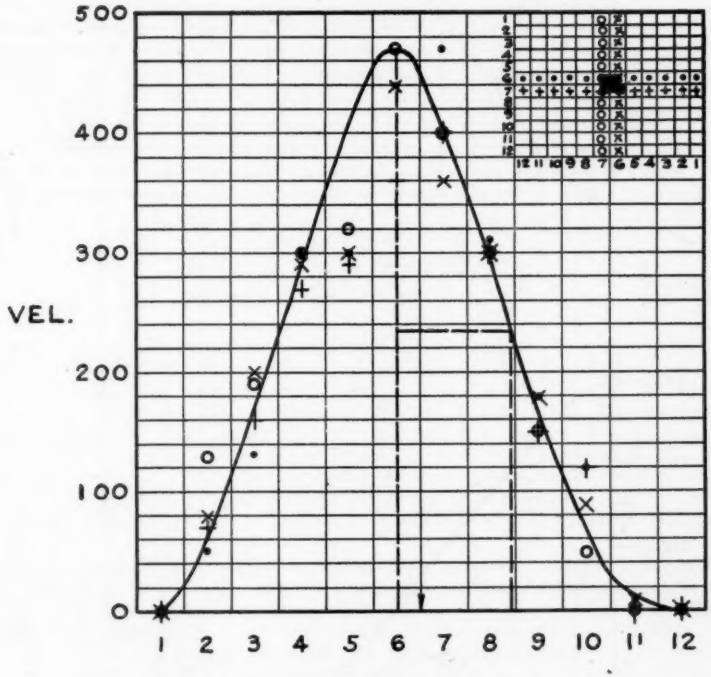

FIG. A. VELOMETER READING *vs.* STATION

(Data for square slot from Fig. 7 in paper)

distance from the slot. When the profiles are symmetrical the maximum velocity occurs along the center line of the slot.

The uncorrected velometer readings taken from Fig. 7 and 8 of the paper are shown in Figs. A and B of this discussion, with smoothed curves set through the data. Both sets of data, although as stated by the authors, show uncorrected velometer readings, indicate considerable non-symmetry which should be explained. Is this non-symmetry inherent or was it caused by the supply conditions to the slot or by incorrect velometer readings? Was the flow steady or were there small pulsations and what effect did they, if existing, have on the velometer readings? Does the velometer read the x component of the velocity?

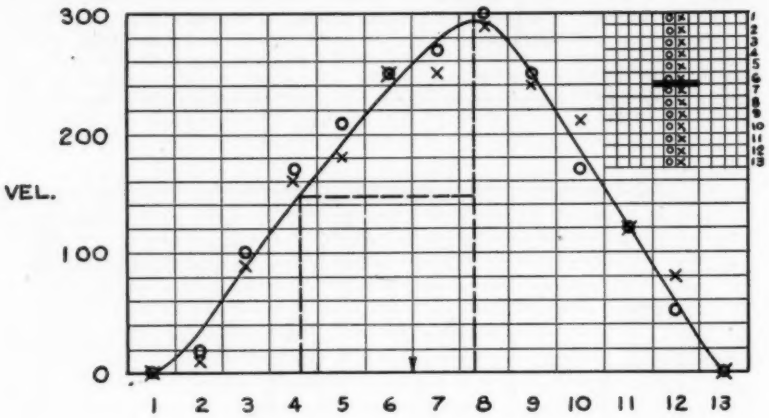

FIG. B. VELOMETER READING *vs.* STATION
(Data for narrow slot from Fig. 8 in paper)

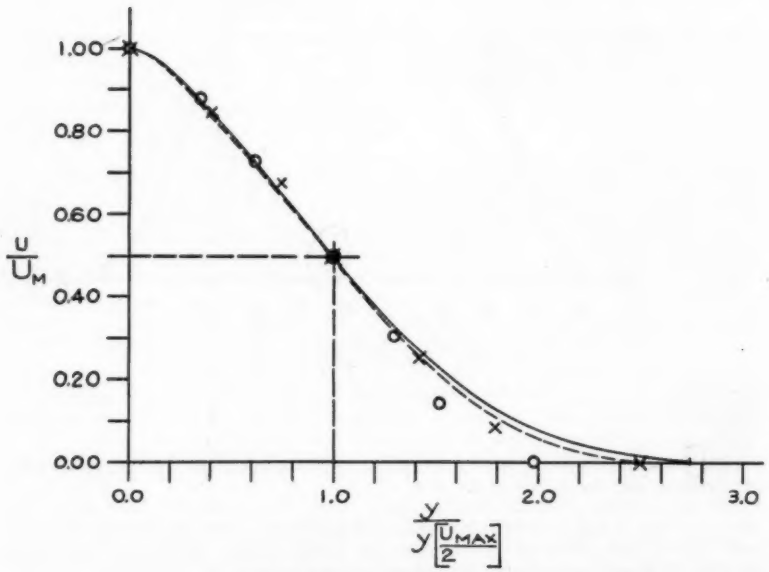

FIG. C. VELOCITY PROFILES

------ Tollmein theory—plane slot—case (*b*)
—— Tollmein theory—rotationally symmetrical case—case (*c*)
x Calculated from smoothed curve of Fig. A.
o Calculated from smoothed curve of Fig. B.

Fig. C shows the theoretical profiles as predicted by the Tollmein theory plotted non-dimensionally in a way which eliminates the empirical constants. The points shown on this figure are calculated points obtained from the smoothed curves of Figs. A and B, which were in turn obtained from the authors' data. The data from the narrow slot show good agreement with case (*b*) of Tollmein while the data from the square slot show good agreement with case (*c*) of Tollmein. The data from the authors' square slot was compared with Tollmein's case (*c*) since at a considerable distance downstream the effect of the slot shape on the profiles, whether square or round, becomes neligible.

No other velocity profiles were given in the paper; however, the writers suggest that an examination be made of the authors' complete corrected velocity profile data to determine how closely the data fit the theory. This can probably best be done by utilizing a method of plotting similar to that of Fig. C.

The authors refer to the *momentum theory* and have gone to considerable trouble to calculate and prove that the *total momentum* (ΣwV^2A) is essentially constant throughout the jet. The momentum theorem is one of the fundamental theorems of fluid mechanics. It is usually obtained by an application of Newton's laws.[22] If correctly applied, the theorem of momentum, being fundamental, will always hold. The theorem of momentum by itself will not furnish much information regarding the mixing action. The quantity which the authors call *total momentum* will obviously be constant provided, (1) the resultant pressure in the x direction on the bounding surfaces of the jet is zero, (2) there are no impressed forces such as gravity, and (3) the flow is stationary. Under these conditions the $\int\int wV^2dA$ will be constant over vertical planes throughout the jet (V is the x component of the velocity). It seems to the writers that the authors have, by showing that the calculated *total momentum* is practically constant, simply proved that these assumptions (1), (2) and (3), are essentially correct.

F. J. KURTH, New York, N. Y.: In air entrainment problems we have found that temperature of the streams has a decided influence on the action. Also the entrainment on the mass of air leaving the outlet will vary considerably. Forces of aspiration are working when part of the energy of the out-flowing air draws the air mass back into the outlet. An outlet that releases an entrainment without aspiration results in a completely different result with respect to aspect ratio, distance and velocity as compared to an outlet that has aspiration. I suggest that future research investigate the fundamental effect of aspiration on air diffusion.

MR. PRIESTER: With reference to Mr. Ashley's comments, we used a plenum and a flat plate outlet more for simplicity's sake, so that we could obtain data on the variables in an easier manner. We realize that more work should be done with a duct approach to an outlet. It was observed that the air stream contracted as it left the sharp edged orifice. We would like to have proved equation 2 in the paper, namely, that the throw is equal to some constant times the velocity times the square root of the dimensions, but we were unable to take enough readings on total throw to complete the entire analytical data. In Table 9, it will be observed that the first item for the square opening is out of line with the rest. This is probably the one figure which Mr. Ashley mentioned and it might possibly be that this figure should have been higher. We took only a very few throw readings. In answer to the comment on counter flow effect, our work was done in that part of the stream close to the outlet, hence the effect of back flow should be negligible. Investigation now being done at the University of Illinois is taking into account total air flow into the space, and data concerning the counter-flow effect should result from this work.

Answering Mr. Zieber, the direct comparison of theory against practice was shown in the slide presented, but was not included in the paper. In other words, we calculated theoretical values after we found from test the average jet angle. A small

[22] Fundamentals of Hydro- and Aero-Mechanics, Prandtl and Tietjens. (McGraw-Hill Book Co., Inc., New York, N. Y., 1934.)

amount of work was done on wide angle grilles, but we did not think it sufficient to add it to this paper. We are not in a position to make a definite statement at the present time regarding velocity and noise, except that we can state that the type or design of grille has a definite effect on noise.

We tested single slots only, so that we do not know the effect of the spacing of two narrow slots or of using two slots as an outlet.

Professor Coogan raised a question concerning whether pulsations were observed. We did observe pulsations, but it was interesting to note the following. This work was carried on over quite a long period of time, and various men took readings. Some men would repeat the work that some of the others had done; but the average entrainment for each outlet, whether obtained by one man or another, agreed, although the reading of the velocity or the velometer reading in each square was not always the same. That is, one man would not get a reading in a certain square that was exactly the same as another man would get running the test at a different time. But the average throughout the whole traverse of the grid was very close.

No. 1205

DETERMINING SOUND ATTENUATION IN AIR CONDITIONING SYSTEMS

By D. A. WILBUR* AND R. F. SIMONS,** TROY, N. Y.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with Rensselaer Polytechnic Institute.

INTRODUCTION

THIS paper reports part of a three-point program initiated by the ASHVE Research Technical Advisory Committee on Sound Control.[1] The basic object is to make possible calculation in advance of installation of the noise levels which will be generated by heating, ventilating and air conditioning equipment. In order to do this it is necessary to know:

1. The amount of noise energy which will be generated by such equipment.
2. Losses which will normally occur as the sound passes through the ventilating duct system.
3. The effect of the conditioned space itself and its noise level upon the residual sound energy from the system.

This report deals with item 2. It describes the development of an experimental technique, and the results of experiments on the first aspect of the problem, the energy losses in straight, unlined ducts of various sizes. Future investigation will deal with sound energy losses in elbows, dividers, changes of area, etc. It is hoped that when the work is completed it will be possible to compute sound attenuation with the same confidence as air flow resistance.

DEVELOPMENT OF MEASUREMENT METHOD

A method which may be relied upon to give complete information must meet the following requirements: (*a*) provision must be made for measuring the attenuation at different frequencies, since in general it varies with frequency; (*b*) the effects of reflection at the ends of the test duct must be minimized

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[1] *Committee Personnel*—J. S. Parkinson, *Chairman;* E. L. Anderson, C. M. Ashley, A. L. Kimball, V. O. Knudsen, R. D. Madison, R. F. Norris, C. H. Randolph, A. E. Stacey, Jr., A. G. Sutcliffe and R. M. Watt, Jr

Presented at the 48th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January, 1942.

in order to separate the attenuation caused by the duct itself from that caused by junctions with other parts of the system; (*c*) provision must be made for exploring the sound level throughout the length of the test duct, for the purpose of determining whether or not the attenuation is the same at all points along the duct, and in order to find the magnitude and phase of any residual standing wave pattern.

In developing a method to meet these conditions several sequences of measurements were made in accordance with the following schedule:

1. Measurements on a small duct (6 in. x 6 in. x 20 ft, 26 gage) terminated in an absorbing conduit.
   a. Single frequency measurements.
   b. Measurements with narrow bands of frequencies.
2. Measurements on a large duct (12 in. x 36 in. x 20 ft, 22 gage).
   a. Duct terminated in an infinite baffle.
   b. Duct terminated in a lumped impedance network providing high absorption at a particular frequency.

APPARATUS

The experimental arrangement for part 1-*a* is shown in Fig. 1. The sound source consisted of a 4-in. cone-type electrodynamic loud speaker *A* driven by a beat frequency oscillator *C*, whose frequency range is 0 to 20,000 cycles per second. Thus condition (*a*) was satisfied. The speaker was mounted in the absorbing baffle box *B* in order to materially decrease the acoustic coupling between the speaker and the air in the laboratory. The reason for thus enclosing the speaker was to prevent variations in the standing wave pattern set up in the room from causing the output of the speaker to fluctuate when it was driven at a single frequency.

Condition (*b*) was met by terminating the duct *D* in the absorbing conduit *E*. This conduit was a wooden tube 20 ft long having the same transverse dimensions as the test duct, and was lined with rock wool whose thickness increased with the distance away from the test duct, as shown in the figure.

The flexibility of the sound level measuring probe system demanded by condition (*c*) was the most stringent requirement to meet. The system shown in the figure was finally considered to be the most satisfactory. The probe element consisted of a diaphragm actuated Astatic crystal microphone element *F* (diameter 2 in., maximum thickness 0.5 in.). This microphone was suspended by small rubber bands from the shielded cable *G* which, together with the turnbuckle *H*, formed an endless belt passing over the pulleys *P*. The microphone was electrically connected to the two conductors of the cable *G* by very fine stranded wire. Thus mechanical vibrations of the cable were quite effectively prevented from reaching the microphone. The longitudinal position of the microphone was determined from the position of the turnbuckle which was read in the linear scale *I*. The end of the cable at the turnbuckle *H* was electrically connected to the input circuit of voltage amplifier *J*, across whose output terminals were connected the copper-oxide voltmeter *V*, and the cathode-ray oscilloscope *O*. Since the microphone element was actuated by the sound pressure, the voltmeter reading at any particular frequency was taken to be proportional to the sound pressure at the point in the duct where the microphone was located. The oscilloscope was used for monitoring purposes: to detect distortion or extraneous sound which would give a spurious reading

on the voltmeter. The small size of the probe system minimized the disturbance of the sound field due to diffraction, and the continuous motion of the microphone permitted the determination of relative sound pressure at any point along the duct.

TEST PROCEDURE

The first procedure with this experimental arrangement consisted of a determination of the longitudinal pressure distribution in the duct. The following steps were executed: the loud speaker was energized at a single frequency and the level maintained constant. The microphone was then moved down the duct, with frequent pauses, at which points the position of the turnbuckle and

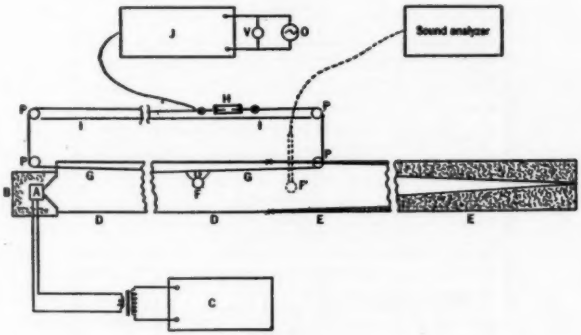

FIG. 1. BASIC ARRANGEMENT FOR DETERMINING ATTENUATION IN SMALL DUCTS

the voltmeter reading were noted. Care was taken to make readings at the points of maximum and minimum sound pressure. This procedure was repeated at many different frequencies. The values of relative sound pressure thus obtained were plotted against the longitudinal position of the microphone as shown in the representative curves of Fig. 2. It should be noted that these curves have two distinct characteristics: a fluctuation (in some cases definitely periodic) superimposed on a constant slope. The fluctuation may be interpreted as being caused by a standing wave system either in the air or in the metal walls of the duct. The constant slope gives a measure of the exponential attenuation (since the relative pressure has been plotted on a logarithmic scale).

It was found that for frequencies greater than 1100 cycles per second the sound pressure at any given longitudinal position varied markedly from point to point in the transverse section of the duct, while for frequencies below 1100 cycles per second, the pressure was quite uniform over any given cross section. This condition is accounted for by the existence of systems of transverse standing waves, which is permitted only for frequencies greater than those whose wave length is twice the largest transverse dimension of the duct. This transverse variation of pressure made the interpretation of curves like

those in Fig. 2 extremely difficult for frequencies greater than 1100 cycles per second. In order to circumvent this difficulty, it was decided to substitute a narrow band of continuously distributed frequencies for the single frequency used before. This procedure would *blur* the standing wave patterns, thus

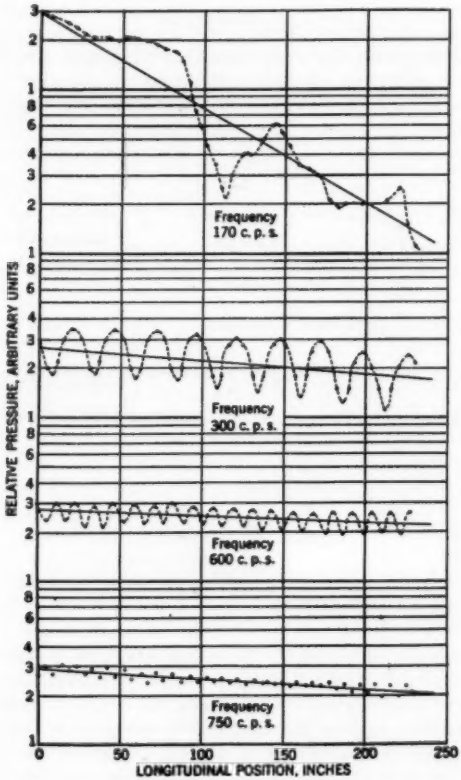

FIG. 2. LONGITUDINAL VARIATION OF SOUND PRESSURE IN 6 IN. X 6 IN. DUCT (SINGLE FREQUENCY METHOD)

giving a more nearly uniform pressure distribution over any particular cross section, while at the same time it would preserve a certain amount of *resolution* with respect to frequencies, so that the variation of attenuation with respect to frequency could still be determined.

This modification was accomplished in the following way: The beat frequency oscillator of Fig. 1 was replaced with a *shot-effect* random frequency generator, consisting of a high gain amplifier equipped with a fast-acting level compression circuit. The output of the system consists largely of amplified

tube noise caused by discontinuities in the electric emission in the first tube of the amplifier. This noise is evenly distributed throughout the frequency spectrum; hence, the output sound signal contains all frequency components which lie within the pass band of the amplifier-speaker combination. The purpose of the compression circuit (or automatic volume control) was to minimize random changes in the output level which would cause the needle of the measuring instrument to fluctuate widely. The copper oxide voltmeter was replaced by a sound analyzer. This instrument selects a narrow band of frequencies (whose central frequency may be varied continuously) and indicates its relative pressure on a voltmeter calibrated directly in decibels. The width of the selected band is a constant percentage of the central frequency over the entire range of the instrument. Some idea of the width of the band may be had from the statement that the sensitivity of the instrument at a frequency ten per cent higher or lower than the central frequency is approximately 10 db below the sensitivity at the central frequency.

With the apparatus thus modified, the probe microphone was temporarily dismounted from its cable, mounted on the end of a small brass tube, and inserted through the side of the conduit *E*, near the test duct, for the purpose of measuring the transverse pressure distribution. This arrangement is shown in dotted lines in Fig. 1, where the microphone is designated F^1. The brass tube contained the microphone cable, and was so mounted as to enable the microphone to be located anywhere in the transverse plane of the conduit. Relative pressure readings were taken over the transverse plane at several frequencies and representative curves are shown in Fig. 3. The variation in level was found not to exceed $\pm$ 1.5 db, which was considered satisfactory for the purpose of making longitudinal runs as was done with single frequencies below 1100 cycles per second.

The microphone was then remounted on the cable as before, and the procedure described for the single frequency measurements was followed. Since many frequencies were now present in the measurement signal, the longitudinal standing wave pattern was blurred and irregular, and there was no longer any reason to search out longitudinal maxima and minima. Points were taken at equally spaced positions, and the relative pressure plotted directly in decibels against longitudinal position, as in the representative curves of Fig. 4. The attenuation may be obtained from the average slope of the curve as was done in the case of the single frequency curves.

The method thus finally selected was satisfactory for the determination of attenuation as a function of frequency for small ducts. There remained the problem of extending it to large ducts. For this purpose a test section whose transverse dimensions were 12 in. x 36 in. and whose length was 20 ft was chosen. An absorbing conduct for the termination of such a duct was considered to be too unwieldy. Therefore, other means had to be used to provide a termination which would absorb a substantial part of the incident sound power. One way of accomplishing this was to terminate the duct in an infinite baffle. Then, for frequencies greater than a critical frequency (determined by the size of the duct opening) most of the energy arriving at the open end of the duct would be radiated into the space beyond the baffle. For the 12 in. x 36 in. duct the critical frequency was in the neighborhood of 200 cycles per second. These considerations were based on the classical analysis of a plane

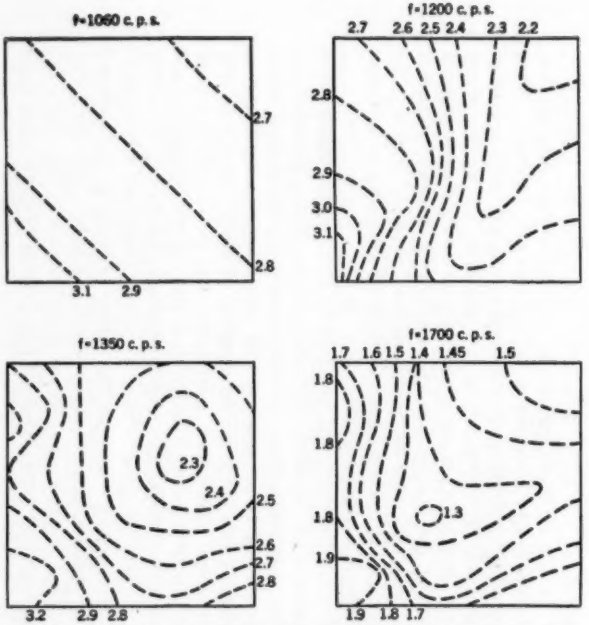

FIG. 3. (*above*) TRANSVERSE DISTRIBUTION OF PRESSURE IN A 6 IN. x 6 IN. DUCT

Dotted lines are sound isobars in transverse section of 6 in. x 6 in. duct. Numbers give relative sound pressure in arbitrary units.

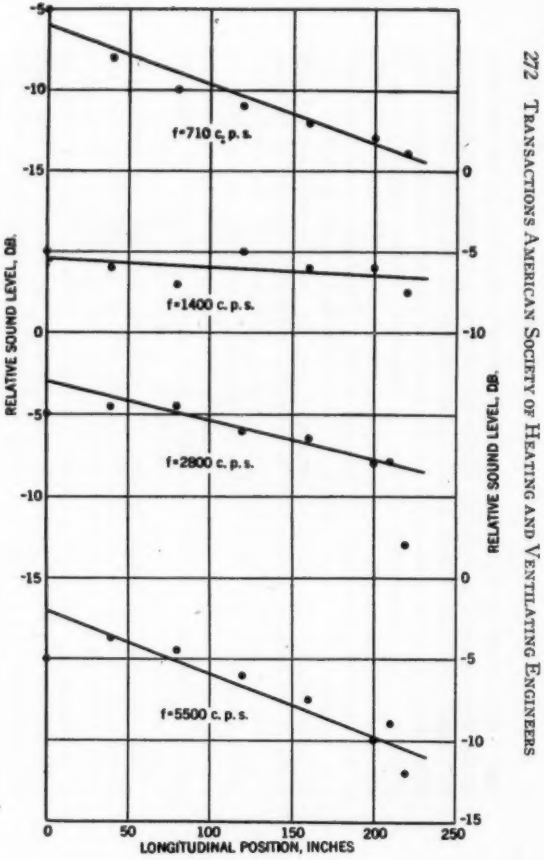

FIG. 4. (*right*) LONGITUDINAL VARIATION OF SOUND PRESSURE IN 6 IN. x 6 IN. DUCT (NARROW BAND METHOD)

radiator vibrating in an infinite baffle. They were later verified by an independent experimental method.

To apply this method, the duct was terminated in an opening in the wall of the laboratory, as shown in Fig. 5. The remainder of the experimental arrangement was similar to that used on the 6 in. by 6 in. duct, except that the 4 in. dynamic speaker was replaced by one of 12 in. diameter. Data were obtained by the procedure outlined before, measurements being made with narrow bands of frequencies to average out the transverse standing waves. Sample curves of relative pressure versus longitudinal position are shown in

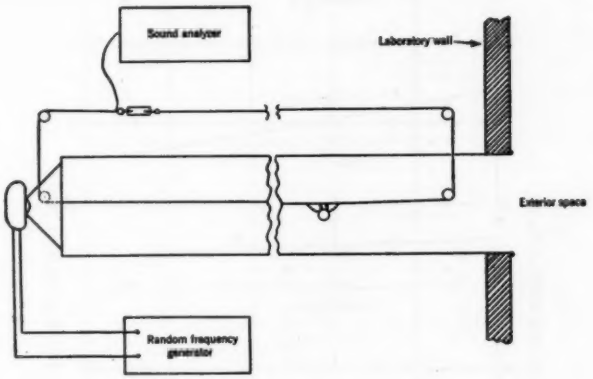

FIG. 5. INFINITE BAFFLE TERMINATION FOR LARGE DUCTS

Fig. 6. Attenuation was again determined from the slope of a median straight line drawn between the points.

In order to check the validity of the assumption that reflection from the baffled end of the duct could be neglected at frequencies greater than 200 cycles per second, and also to extend the range of measurements down to lower frequencies, recourse was had to a method of producing a controllable terminating impedance developed by Bolt and Brown.[2] In broad outline, this method consists of terminating the duct in a network of lumped acoustical constants whose impedance may be made to have any desired value at any desired single frequency, within a wide range of both variables. In particular this network may be made to *match* the duct impedance, so that negligible reflection occurs at the chosen frequency. The particular form of network chosen for this investigation is shown in Fig. *7A*, and its equivalent electrical network in Fig. *7B*. The rock wool partition in a series of combination of inertance and resistance, while the volume enclosed by the partition and the rigid wall at the duct end is an acoustic capacitance. At some frequency the net reactance will be zero, and if the remaining resistance is equal to the characteristic impedance of the duct the reflection from the partition into the duct will

[2] Variable Boundary Impedance for Room Acoustics Investigation, by Richard A. Bolt and Richard L. Brown, (*Journal of the Acoustical Society of America*, Vol. 12, July, 1940).

vanish. Since this termination is essentially a resonant system, its behavior is extremely sensitive to changes in frequency in the neighborhood of resonance where the reactance is zero. For this reason, it was decided to work with single frequencies, and accordingly, the random frequency generator was re-

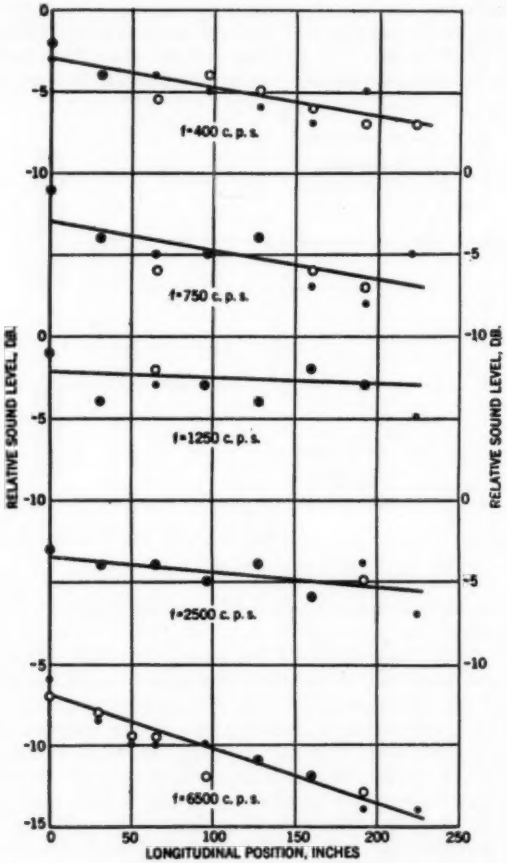

FIG. 6. LONGITUDINAL VARIATION OF PRESSURE IN 12 IN. X 36 IN. DUCT (NARROW BAND METHOD)

placed with a primary oscillator, chosen for its frequency stability. Proper values of the partition thickness and air chamber volume were determined by trial since the effective resistance and inertance could not very well be calculated in advance. Also, during this investigation an alternative method of microphone support was tested. This consisted of hanging the microphone

from two vertical supports mounted on a car equipped with rubber tired wheels. This car was driven through the duct by an endless cord passing over pulleys as in Fig. 1. The microphone cable then lay on the floor of the duct, being dragged behind the car. In this way the endless belt did not have to support the weight of the microphone, which was an advantage. On the other hand, the volume occupied by the car was many times greater than that of the microphone alone, and hence the diffraction effects were probably more serious; however, these could be neglected at the lower frequencies. Representative data taken with this arrangement are shown in Fig. 8, from which attenuation could be evaluated at frequencies below 200 cycles per second. Attenuation for frequencies above this value agreed with values obtained from

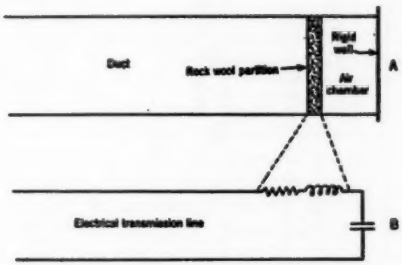

FIG. 7. LUMPED IMPEDANCE TERMINATION FOR LARGE DUCT

data taken with the duct terminated in the infinite baffle, thus justifying this method at the higher frequencies.

CONCLUSIONS

From the results of the tests previously described, some general conclusions about the method of measurement may be drawn.

1. Attenuation depends upon the frequency of the sound, and hence measurements must be made at different frequencies throughout the audible range.
2. Where the attenuation is small, measurements must be made throughout the length of the test duct. The attenuation may then be calculated from the slope of a median straight line drawn through the *center of gravity* of the points. Figs 2, 4, and 6 show that under this condition, it is impossible to attempt to evaluate the attenuation merely from the ratio of the sound pressure at one end of the duct to that at the other end.
3. For small ducts an absorbing conduit affords a satisfactory termination, while for large ducts an infinite baffle satisfies the requirements.
4. A narrow band of frequencies is to be preferred to a single frequency when determining attenuation at a particular frequency. The loss in frequency resolution is more than compensated for by the elimination of confusion due to transverse standing waves.
5. The method previously described meets the conditions established in the first paragraph. When working with systems having low attenuation, some method such as this one, which satisfies these conditions, must be used to obtain reliable results. Also, this method may be used to check the reliability of simpler methods which may be used when the attenuation is larger.

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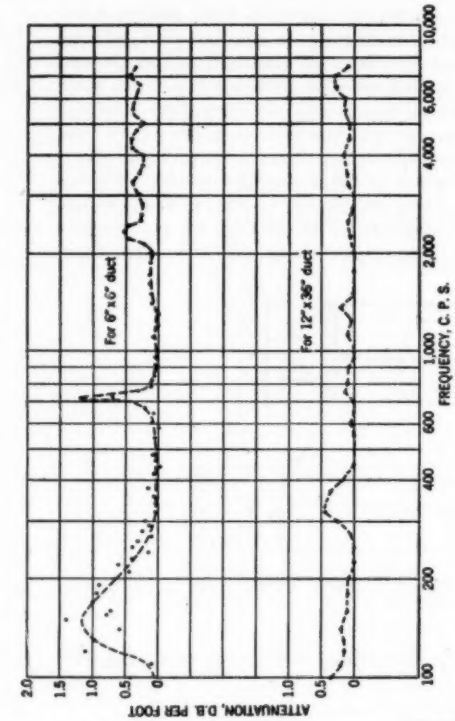

FIG. 9. ATTENUATION CHARACTERISTICS OF STRAIGHT UNLINED DUCTS

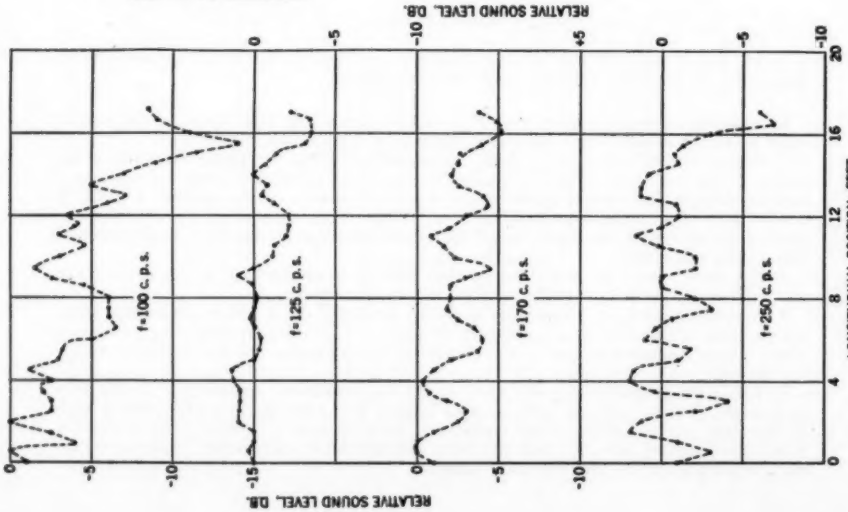

FIG. 8. *(left)* LONGITUDINAL VARIATION OF SOUND PRESSURE IN 12 IN. x 36 IN. DUCT (SINGLE FREQUENCY METHOD)

Attenuation in Unlined Ducts

If the attenuation per unit length in a duct is the same at all positions along the duct, the sound pressure, and hence intensity, in a progressive wave may be given by the equation

$$p = p_0 \epsilon^{Qx}$$

where

p = pressure at point x.
p_0 = pressure at arbitrary origin.
x = position along the duct.
Q = attenuation constant, in general a function of frequency.

When the logarithm of p is plotted against position, this equation reduces to a straight line, whose slope is Q. In Figs. 2, 4 and 6 a median straight line has been drawn between the points, thus averaging out the contributions due to standing waves. In no case is there evidence that the median line is curved; thus there is every indication that the attenuation may be specified by a single constant at each frequency.

The values of Q, in decibels per foot, as determined from the slopes of these median lines, are shown plotted against frequency in Fig. 9. The curve for the small duct shows that the attenuation is negligible below 2000 cycles per second except for two regions, one at 700 cycles per second and the other at 150 cycles per second. It was found that in these regions the walls of the duct vibrated as panels, thus radiating considerable power into the space surrounding the duct and, therefore, preventing it from being transmitted along the duct interior. Above 2000 cycles per second there is some measurable attenuation which did not seem to be associated with panel resonance. For the large duct the attenuation was everywhere negligible except in the regions 300-400 cycles per second. These frequencies were likewise associated with a group of panel resonances. On both the large and small ducts these panel resonances could be easily seen when fine sand was sprinkled on the top surface of the duct. If the frequency of the sound were continuously varied, the sand remained undisturbed until one of the resonance regions was entered. It then was set into motion, and if the sound were maintained, would pile up in the regions where the motion of the duct wall was least, thus defining the dynamic deformation pattern of the walls. These patterns were stable and reproducible, but highly irregular.

Since the same type of result was obtained with these two sizes of ducts, it is fair to assume that they are representative of all ducts whose sizes are intermediate between them. The general conclusion then is that for straight unlined ducts the attenuation is less than 0.2 decibels per foot, aside from certain narrow frequency regions where panel resonance may occur, and except for high frequencies in very small ducts.

DISCUSSION

C. M. ASHLEY, Syracuse, N. Y.: I think one of the most important contributions of the paper is in the technique involved in the test. For those who have done any work of this sort, they must realize that there are serious technical problems in determining the attenuation, and I believe this work has gone quite a long way toward exploring the possibilities in that regard.

A second point which I would like to suggest is that I believe the work might be expanded by developing the principle that practically all of the sound loss, the

LONGITUDINAL POSITION, FEET

attenuation, is due to transmission through the panels of the duct. Since duct panel transmission is fairly well known, I believe it would be possible to predict the effect for different size ducts, and different gauges of sheet metal with reasonable accuracy.

I might say for the committee that it is hoped that this work can be carried on into different types of connections, such as outlets, in sequence, and elbows, splitters, and various other things which are believed to have a very substantially larger attenuation than the duct itself.

H. K. KUNEN, New York, N. Y.: I believe one important point which should be investigated by the committee is the effect of air flow on sound attenuation in ducts. No doubt when air flows through any duct or conduit a sound wave is generated, due to the turbulent flow which creates currents. The sound generated should have an appreciable effect on the natural attenuation.

We have conducted a few experiments on various methods of sound control, together with the effect of attenuation in ducts, and we found that by having air flow through the duct at various velocities, results varied appreciably.

MR. SIMONS: In connection with Mr. Ashley's suggestion, mentioning the desirability of extending the method to the effects of elbows and take-offs and splitters, etc., that is now under consideration. We are engaged in investigating the effect of take-offs and of the ratio of the take-off to the main duct, one of the important parameters.

As far as the effect of air flow is concerned, that certainly is an important part of the problem. Obviously a method like this, involving an endless belt which has to be taken in and out of the duct, could not be used very easily where you have air moving through the duct; but you could use this method to check a second method which involves a stationary setting of the microphone in the moving air stream, then this second method could be used to measure the attenuation when air is flowing through the duct.

No. 1206

THE PERFORMANCE OF STACK HEADS EQUIPPED WITH GRILLES

By D. W. NELSON,* D. H. LAMB** AND G. E. SMEDBERG,*** MADISON, WIS.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with University of Wisconsin.

INTRODUCTION

THE present report concerns the influence of grilles when placed on the stack openings. A previous paper [1] reported the results with plain openings; that is, without grilles. The same test setup as used in the previous tests was used in the present series, and is shown in Fig. 1. The apparatus consists mainly of a fan, a long 12 in. duct in which a 5 in. rounded approach nozzle was located, and a plenum chamber on which a 14 in. x 6 in. stack 8 ft high was placed.

PROCEDURE

The same gages and instruments were used with the exception that an improved directional indicator as shown in Fig. 2 was made and found to be more convenient. With each stack head and grille combination, observations were made of the direction and velocity at the center of each of nine equal areas of the face, the direction at the four sides of the air stream 12 in. out from the face, the distance from the bottom edge of the face up to the location at which reversed flow began, the static pressure at the base of the stack head and the pressure readings at the flow nozzle for volumetric determinations. Previous to each test calculations were made to determine the proper gage settings for velocities in the stack of 200, 500, 800, 1100 and 1400 fpm. These speeds correspond to a Reynolds' number based on the stack area of 14,400, 36,000, 57,500, 79,100, and 100,600, respectively.

The grilles used were of diverse patterns, as shown in Fig. 3, and were chosen for having considerable influence on the air stream. A description of the grilles and the free areas when used with each of the stack heads Nos. 4,

* Associate Professor of Mechanical Engineering, University of Wisconsin. MEMBER of ASHVE.
** Formerly Research Assistant, University of Wisconsin, 1940-41.
*** Research Fellow, University of Wisconsin, 1940-42.

[1] The Performance of Stack Heads, by D. W. Nelson, D. H. Krans and A. F. Tuthill, (ASHVE TRANSACTIONS, Vol. 46, 1940.)

Presented at the 48th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January, 1942.

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5, and 6 is given in Fig. 4. This figure also shows the dimensions for these three square throated stack heads.

The direction of air flow in each case was from right to left of the grille section, as shown in Fig. 4. The grilles listed as No. *7B* and *7D* were grille No. 7, used as shown in the upper and lower section, respectively, for this

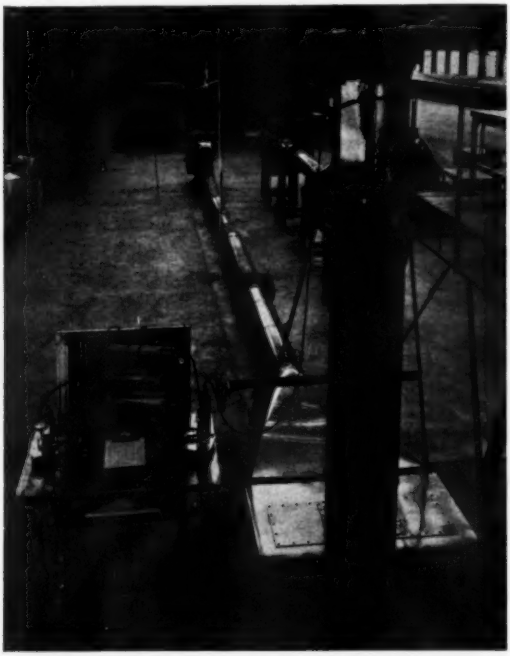

FIG. 1. VIEW OF APPARATUS

grille, and grille No. *7A* and *7C* for the air flow in the same direction, but with the grille inverted to direct the air upward.

Table 1 lists the symbols used to designate the various stack head and grille combinations on the curves of Figs. 8, 9, 10, and 17.

DISCUSSION OF RESULTS

Discussion of the performance of the various stack heads with grilles will first consider velocities and negative flow areas, then directional effects, and finally static pressures at the base of the stack head.

Velocities and Negative Flow Areas

Without a grille, the tendency is for the highest velocity to be in the upper third of the stack head face and for the reverse flow areas to be large, being

largest with stack head No. 6 and smallest with stack head No. 4. In Figs. 5, 6, and 7, the velocities and directions of the air streams are shown for stack velocities of 500 and 1100 fpm for stack heads Nos. 4, 5, and 6, respectively. In these figures the variations in velocity and in the area of negative flow are shown graphically. Without a grille, stack head No. 6 with a 14 in. x 12 in. opening has a reverse flow area which occupies the entire lower half of the face. With high average velocities and grilles of low free area, the velocity in the top third of the face is slightly lower than the average velocity, while with grilles of high free area the velocity in the top third is higher than the average velocity.

From Fig. 8, in which the negative flow area in per cent of stack head area is plotted against the per cent free area of the stack head, it is apparent that

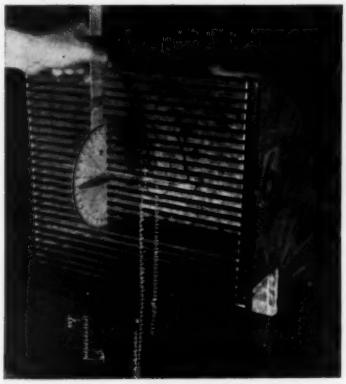

FIG. 2. VIEW OF DIRECTIONAL INDICATOR

there will always be an area of negative flow unless the free area is reduced by either restricting the stack head outlet or by use of a high resistance grille such as Nos. 5 and 6. It will also be noted from this same figure that each of the stack heads have no negative area when a grille of 25 per cent free area is used. With a free area greater than this value, the negative area increases. This effect is most pronounced with stack head No. 6, and least pronounced with stack head No. 4. In Fig. 9, the net discharge area in per cent of stack head area is plotted against the per cent free area of the grille. It will be noted on Fig. 8 that for the stack heads tested, a grille with a free area of 25 per cent results in the spreading of the air over the complete face of the grille; that is, the complete removal of the negative flow area.

It will be seen from the flow patterns of Figs. 5, 6 and 7 that a grille of low free area, as grilles Nos. 5 and 6, results in a more uniform flow over the face than that obtained with one of large free area or one with no grille. However, the velocity is very high at the face due to the low free area, and there is little or no negative area at the bottom of the grille. With grille

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No. 7 having deflecting fins, no difference in negative flow area was found, whether the air was directed upward or downward, as shown in Figs. 5, 6, and 7. The several tests indicate that as the stack head height is increased, the per cent free area used decreases. This would approach zero per cent at an infinite stack height. Extension of such a curve toward zero stack head height indicates that with a face height of approximately 4.5 in. or smaller and a stack depth of 6 in., no negative flow area would occur. With the addition of grilles, discharge over the entire face area can be secured for stack heads with face heights of more than 4.5 in.; the less the free area, the greater will be the face height without negative flow. However, the net discharge area used has not been increased by this procedure. From this it appears that there is an optimum face height for each type of stack head with

FIG. 3.—VARIOUS TYPES OF GRILLES TESTED

a certain stack depth in order that no negative flow will be present. Figs. 8 and 9 are related to this discussion.

Directional Effects

In Fig. 11 a graphical summary of the angles of discharge for the air stream in the upper third of the face opening is shown for various grille and stack head combinations at 500 and 1100 fpm stack velocities. From this figure and Tables 2, 3 and 4 it will be noted that at each combination of stack head and grille the velocity in the stack does not affect the angle of discharge appreciably. For example, the angle for the stack head No. 4 without a grille is approximately 7 deg up for all stack velocities used. This indicates that the velocity in the stack is not a major factor, and that the type of stack head, its dimensions, and the type of grille used are the deciding factors as regards the angle of discharge. In general, the upward tilt of the air stream increases with face height. This angle is greatest for stack head No. 5, and least for stack head No. 4. For stack head No. 6, with the greatest face height, this trend does not hold closely due to the guiding effect of the greater face to throat dimension. The angle without grilles for stack head No. 6 was about 18 deg, as against 27 deg for No. 5 and 8 deg for No. 4. Fig. 11 shows the pronounced directional effect of grille No. 7 when it is turned four different ways. The stack heads without a grille and with grille Nos. *7A* and *7C* impart a definite upward direction to the air stream, grilles *7B* and *7D* give a downward direction, and grilles 5 and 6 also give a downward deflection, while the other grilles give a direction closer to horizontal. Even though

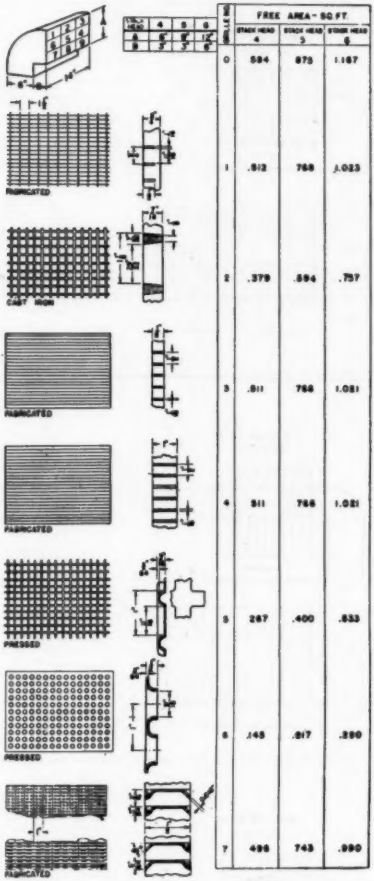

FIG. 4. CONSTRUCTION AND FREE AREAS OF GRILLES AND STACK HEADS TESTED

stack head No. 5 without a grille has the highest angle of discharge, and the No. 4 the lowest in most every case, this is not true when these stack heads are used in combination with the highly directional grille No. 7, which gives nearly the same angle of discharge for all three heads.

In Fig. 10 the direction of the angle of flow is plotted against the per cent of free area of the stack head, and for all grilles except No. 7, which has directional construction. It will be noted that the lower the free area of the grille in per cent of the stack head area, the lower is the angle of discharge.

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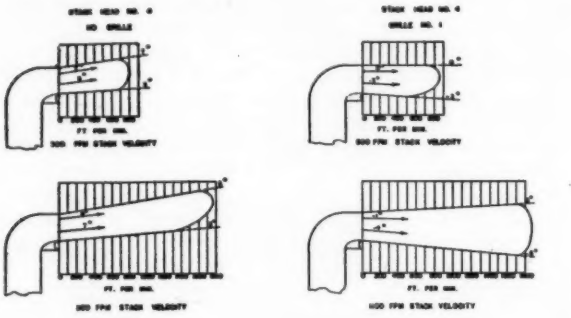
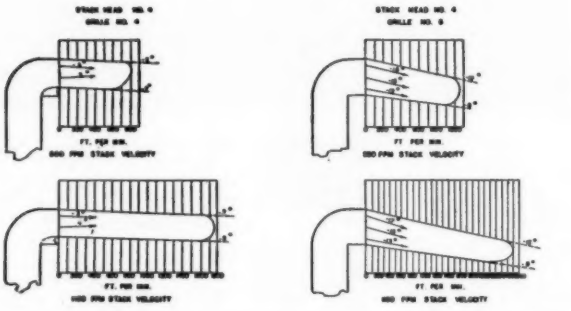
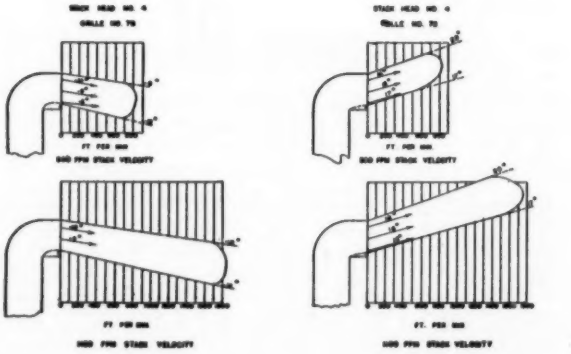






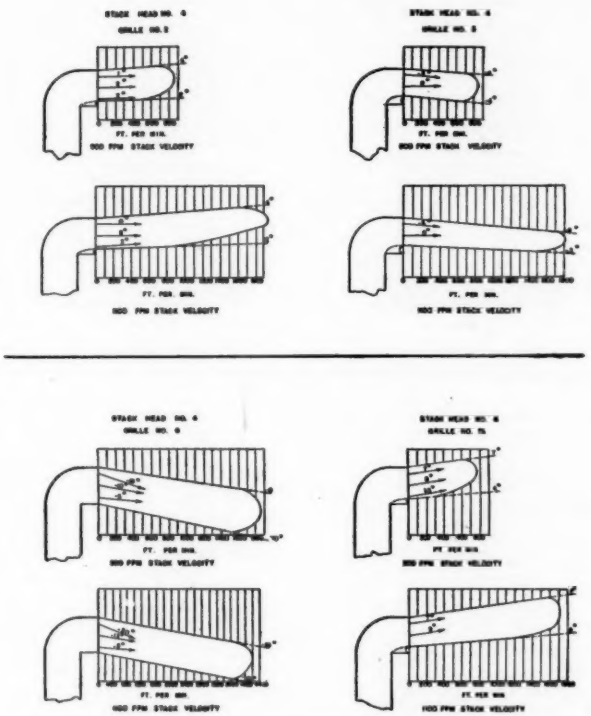
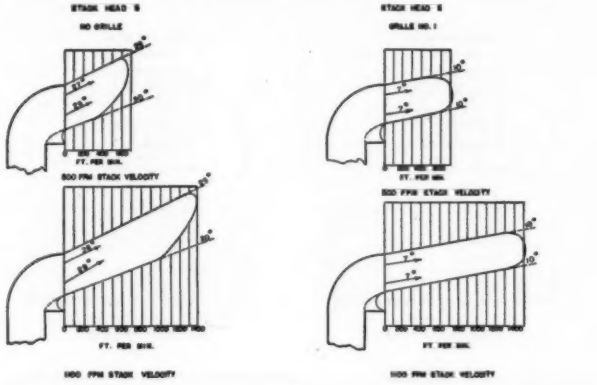
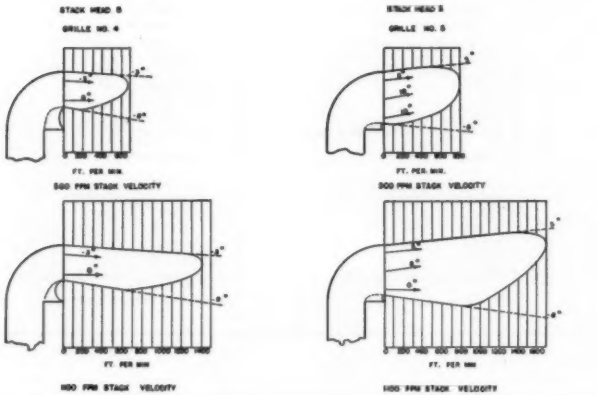
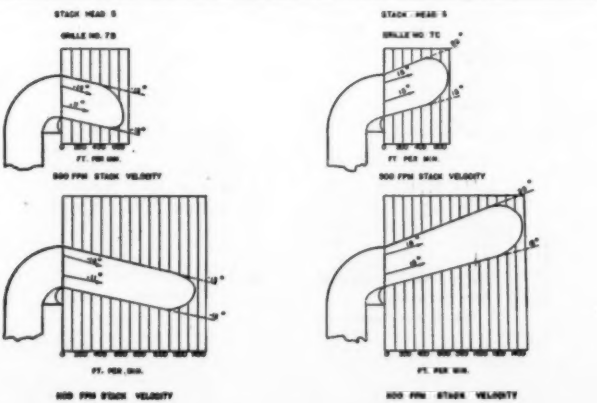


FIG. 5. VELOCITY AND DIRECTIONAL DIAGRAMS FOR VARIOUS GRILLES MOUNTED ON STACK HEAD NO. 4


STACK HEAD S
NO GRILLE
FT. PER MIN.
500 FPM STACK VELOCITY
FT. PER MIN.
1000 FPM STACK VELOCITY
STACK HEAD S
GRILLE NO. 1
FT. PER MIN.
500 FPM STACK VELOCITY
FT. PER MIN.
1000 FPM STACK VELOCITY


STACK HEAD S
GRILLE NO. 4
FT. PER MIN.
500 FPM STACK VELOCITY
FT. PER MIN.
1000 FPM STACK VELOCITY
STACK HEAD S
GRILLE NO. 5
FT. PER MIN.
500 FPM STACK VELOCITY
FT. PER MIN.
1000 FPM STACK VELOCITY


STACK HEAD S
GRILLE NO. 7S
FT. PER MIN.
500 FPM STACK VELOCITY
FT. PER MIN.
1000 FPM STACK VELOCITY
STACK HEAD S
GRILLE NO. 7C
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500 FPM STACK VELOCITY
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1000 FPM STACK VELOCITY

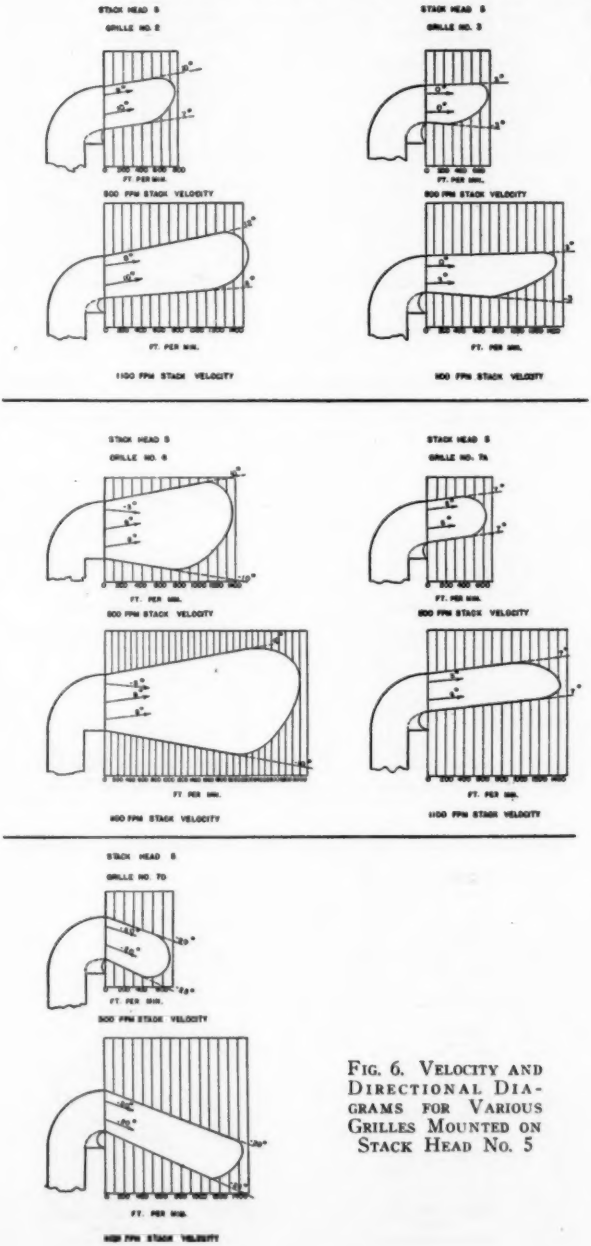
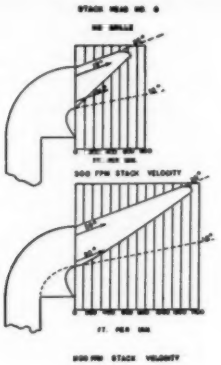
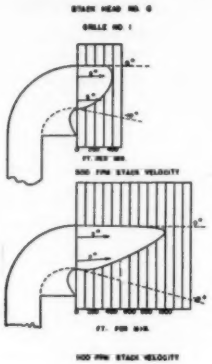
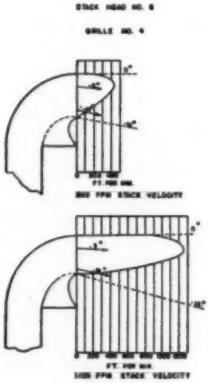
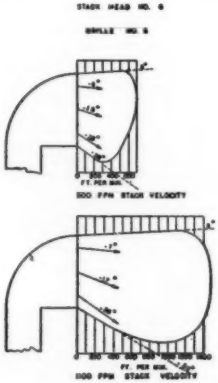


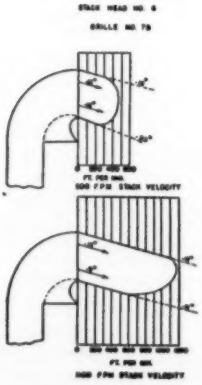
FIG. 6. VELOCITY AND DIRECTIONAL DIAGRAMS FOR VARIOUS GRILLES MOUNTED ON STACK HEAD NO. 5

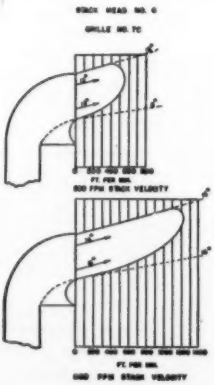












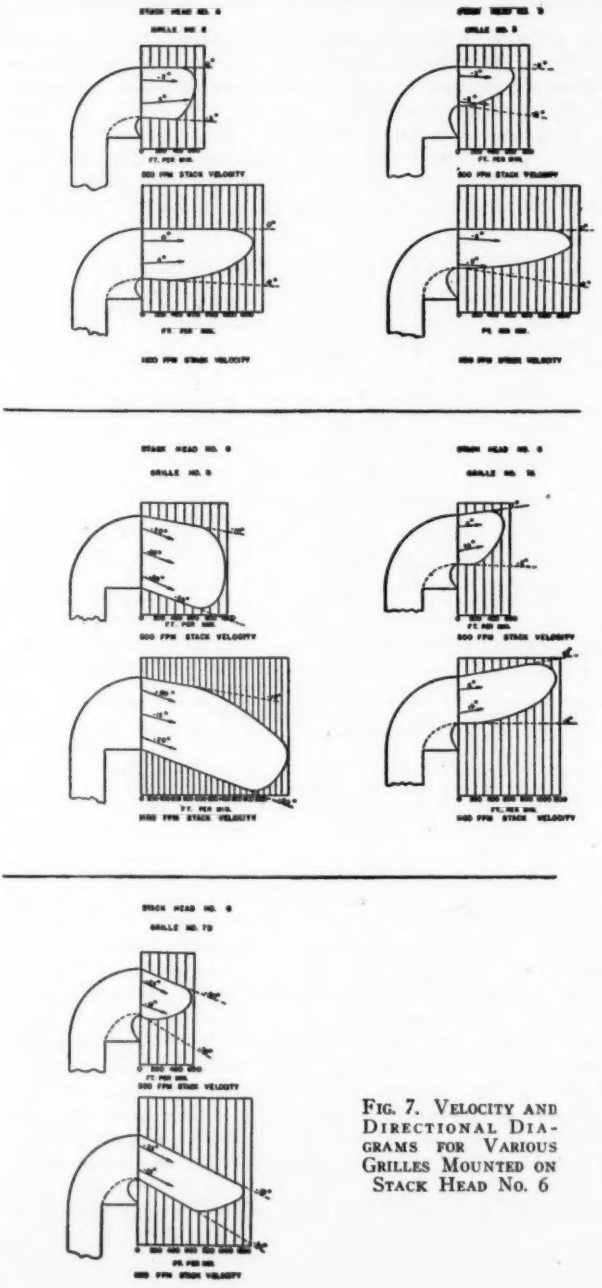


FIG. 7. VELOCITY AND DIRECTIONAL DIAGRAMS FOR VARIOUS GRILLES MOUNTED ON STACK HEAD NO. 6

The non-directional grille No. 6 on stack head Nos. 4 and 6 resulted in a downward deflection of the air stream in the top third of the face opening of about 20 deg. Fig. 10 shows, in general, the greatest downward deflection for grilles on stack head No. 4, intermediate deflections for stack head No. 6, and least deflection for No. 5. From inspection of Figs. 5, 6, 7, and 10, it will be noted that the high resistance grille Nos. 5 and 6 give a definite downward direction to the air stream. The one exception to this statement is with grille No. 5 on stack head No. 5, which lowers the angle from an upward 27 deg to an

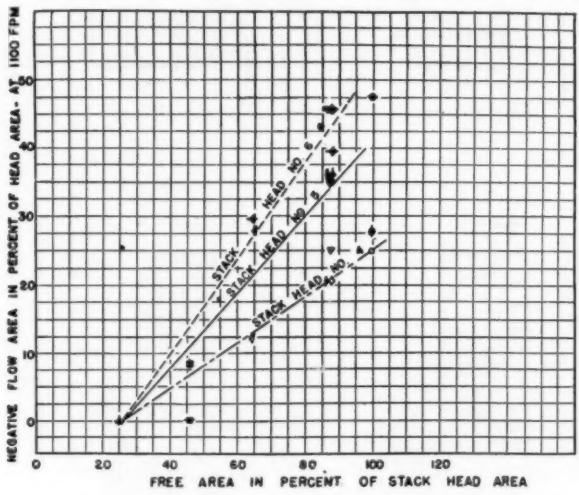


FIG. 8. NEGATIVE FLOW AREA IN PER CENT OF HEAD AREA PLOTTED AGAINST PER CENT OF FREE AREA OF HEAD FOR STACK HEADS NOS. 4, 5, AND 6

upward of 5 deg. This downward deflection seems to be caused by the crowding of the air to the top of the stack head as it makes the turn, and the effect of the high resistance grille in shutting off complete escape at the top of the face opening. Therefore, a considerable quantity of the air stream must escape through grille openings in the lower parts of the face, and the air takes a downward path from the upper part of the stack head to this location. It is therefore leaving the grille openings at a lower angle than when escaping through the unrestricted stack head without a grille or the less restricted face with a low resistance grille. This downward tilt is associated with low negative flow area as well as with high resistance of the face opening. In Figs. 12*a* and 12*b* are shown two photographs taken in an air flow analyzer which indicate this behavior. In Fig. 12*a*, a stack head somewhat similar to No. 6 is shown without a grille, and in which the crowding of the air stream at the top of the stack head is noticeable. In Fig. 12*b*, when the free area of a stack head somewhat similar to No. 4 has been reduced by the addition of

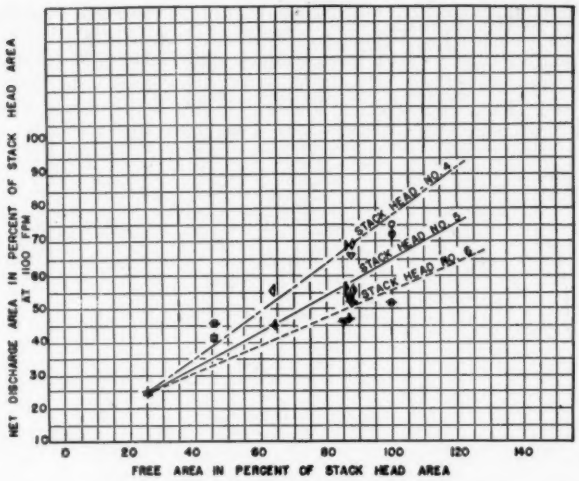

NET DISCHARGE AREA IN PERCENT OF STACK HEAD AREA AT 1100 FPM
FREE AREA IN PERCENT OF STACK HEAD AREA
STACK HEAD NO. 4
STACK HEAD NO. 3
STACK HEAD NO. 2
STACK HEAD NO. 1

FIG. 9. NET DISCHARGE AREA IN PER CENT OF STACK HEAD AREA PLOTTED AGAINST PER CENT FREE AREA IN STACK HEAD

grille elements at the face, the angle of the air stream has been lowered in the process of spreading it more uniformly over the face. With grille elements having less horizontal guiding length and less free area, greater downward deflection would be expected.

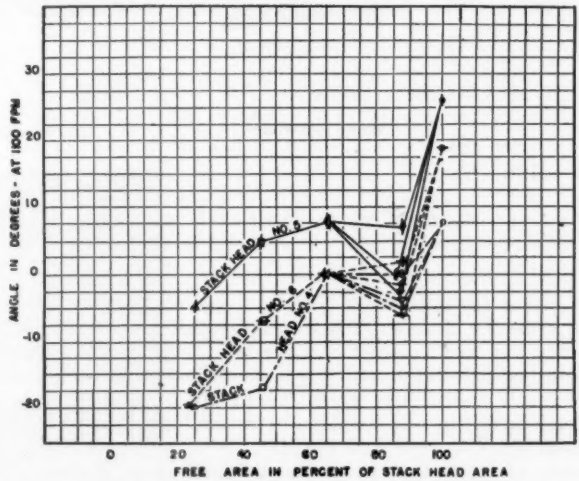

ANGLE IN DEGREES - AT 1100 FPM
FREE AREA IN PERCENT OF STACK HEAD AREA
STACK HEAD NO. 3
STACK HEAD NO. 2
STACK HEAD NO. 4
STACK HEAD NO. 1

FIG. 10. DISCHARGE ANGLE IN DEGREES PLOTTED AGAINST PER CENT FREE AREA OF STACK HEAD

Figs. 13*a* to 13*f* show the discharge from five of the stack head and grille combinations with the aid of smoke. The stack head No. 4 without a grille is shown in Fig. 13*a*, in which the area of negative flow and the upward tilt to the air stream are visible. Figs. 13*b* and 13*c* show stack heads Nos. 5 and 6 also without grilles. The negative flow area in No. 6 covers the whole lower half of the stack face area. These photographs show the relative upward tilts

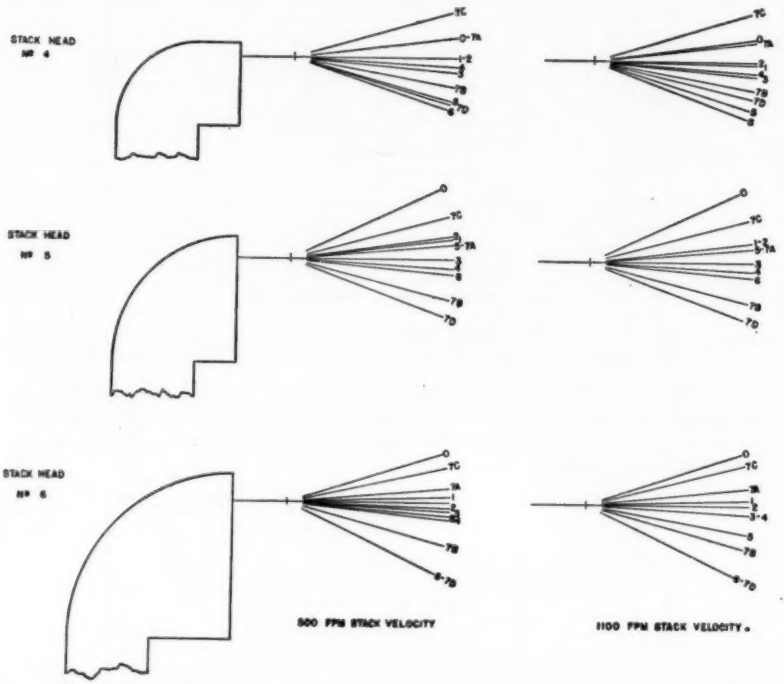


FIG. 11. ANGLE OF DISCHARGE FOR VARIOUS GRILLES AND STACK HEAD COMBINATIONS AT 500 AND 1100 FPM STACK VELOCITY

of the air stream for stack heads Nos. 4, 5, and 6. Fig. 13*d* of stack head No. 6 with grille No. 2 shows some of the negative area still present, but the angle of flow has been lowered considerably. The higher resistance grille No. 5 mounted on the stack head No. 6 is shown in Fjg. 13*e* to have eliminated the negative area and to have given the air stream a downward tilt. The downward tilt is still more pronounced with the high resistance grille No. 6 on stack head No. 6 as shown in Fig. 13*f*.

A summary of results for the various grilles used on stack head No. 4 is shown in Table 2, on stack head No. 5 in Table 3, and on stack head No. 6 in Table 4. Face velocities in Column 3 were obtained by multiplying the

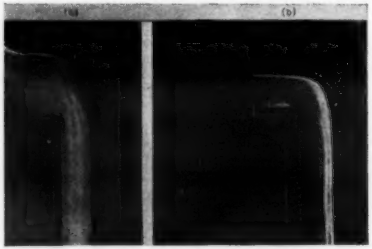


FIG. 12a-b. (*above*) AIR FLOW IN STACK HEADS AS DETERMINED IN FLOW ANALYSER

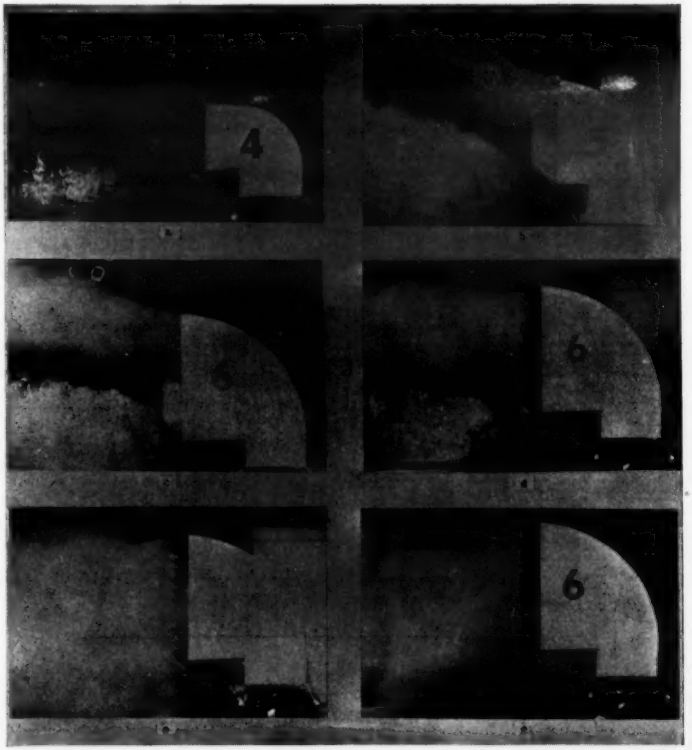
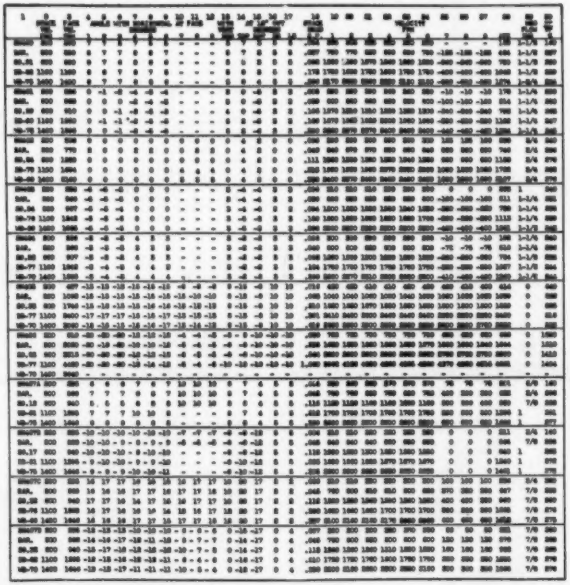


FIG. 13a-f. (*right*) PHOTOGRAPHS OF AIR STREAM LEAVING STACK HEADS

TABLE 1. SYMBOLS USED ON CURVES FOR STACK HEAD AND GRILLE COMBINATIONS

GRILLES	STACK HEADS		
	4	5	6
0	○	⌽	⊖
1	◇	⌖	⊕
2	◁	⌖	⊣
3	▷	⌖	⊢
4	▽	⍒	⊽
5	□	⌖	⊟
6	△	⍋	⊜
7A	◒	⌖	⊖
7B	◓	⌖	⊖
7C	◑	⌖	⊖
7D	◐	⌖	⊖

TABLE 2. RESULTS OF TESTS ON GRILLES WITH STACK HEAD NO. 4



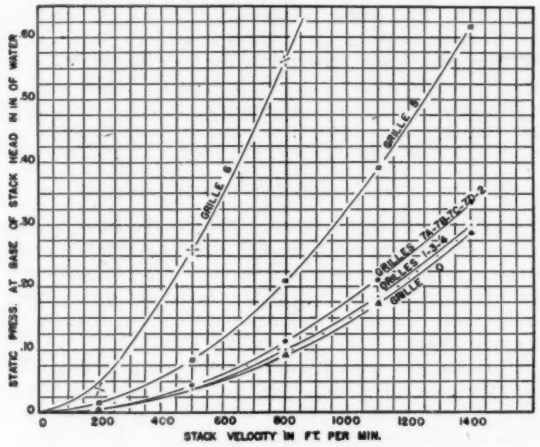


FIG. 14. VARIATION OF STATIC PRESSURE AT BASE OF STACK HEAD NO. 4 WITH VARIOUS GRILLES

stack velocities by the ratio of stack area to the free area of the grille (Fig. 4) on a face opening. Negative values in Columns 4 to 12 indicate downward tilt of the air stream and in Columns 19 to 27 reverse flow.

Static Pressure Effects

In Figs. 14, 15, and 16 the static pressure at the base of the stack head for stack head Nos. 4, 5, and 6, respectively, are plotted against the stack velocities in feet per minute. From these curves it will be noted that the highest static pressures are found with grille Nos. 5 and 6, while for grille Nos. 1, 2, 3, 4, and 7 the static pressures are only slightly above those for stack heads without grilles. These curves indicate that as the face area is decreased, the static pressures at the base of the stack head are increased, and for any individual stack head the results show that the highest static pressures occur when the stack head is equipped with a grille of the least free area. Grille openings are essentially orifices; therefore, in addition to free area there is the element of coefficient of discharge influencing static pressure at the base of the stack head and grille combination at a given air volume. Pressed grille Nos. 5 and 6 are quite similar as to orifice action. Either one would have a higher coefficient of discharge and lower resistance if used in a reversed direction. The cast-iron grille No. 2, which might be expected to have a high resistance did not have, because of a comparatively high free area and a nozzle-like shape from the draught of the molding pattern which would result in a high discharge coefficient. With grille No. 7, which has a definite directional effect, the static pressure was not increased when this grille was reversed to deflect the air downward rather than upward. In Fig. 17 the static pressures at the base of the stack head are plotted against the negative flow areas in inches. The highest negative flow areas result with stack heads without grilles which are

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also accompanied by low static pressures. The lowest negative flow area results with the grilles having the highest resistance. As the free area of the grille decreases, the static pressure at the base of the stack head goes up very rapidly. The static pressure at the base of the stack head reduces from the highest with stack head No. 4, to the lowest with stack head No. 6 with all the grilles tested.

CONCLUSIONS

More uniform flow was found over the face of the stack head with the use of grilles having high resistance than with those having low resistance. Low negative flow area is associated with low face areas of plain stack heads or with the use of high resistance grilles. Reduction of negative flow area results in higher face velocities and reduction of negative flow area by use of high resistance grilles is more pronounced with high face heights.

The angle of discharge from stack heads with various grilles is not influenced appreciably by the velocity in the stack, but is affected by the type of stack head and grille. In general, the angle of discharge is lower with decreased stack head face height and with smaller free area of the grille. With low free areas of grilles negative angles of discharge were obtained. Pressed grilles gave angles of discharge below the horizontal, whereas grilles with horizontal bars resulted in discharge more nearly horizontal.

With decrease in net face area, the static pressure increases slowly at first while the negative flow area is decreasing to zero; with further decrease of free area the static pressure at the base of stack head goes up rapidly. Pressed grilles had the lowest free areas which, together with low coefficients of discharge, resulted in the highest static pressures in all cases.

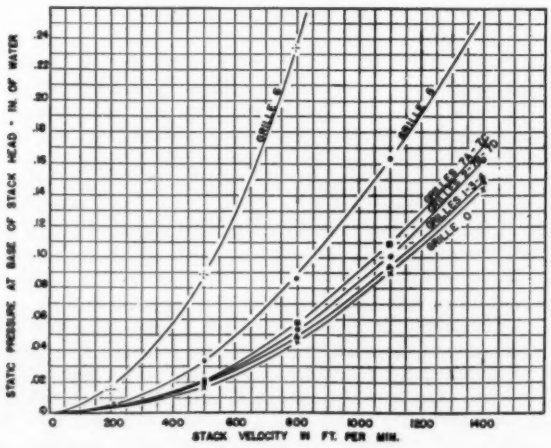


FIG. 15. VARIATION OF STATIC PRESSURE AT BASE OF STACK HEAD NO. 5 WITH VARIOUS GRILLES

TABLE 3. RESULTS OF TESTS ON GRILLES WITH STACK HEAD No. 5

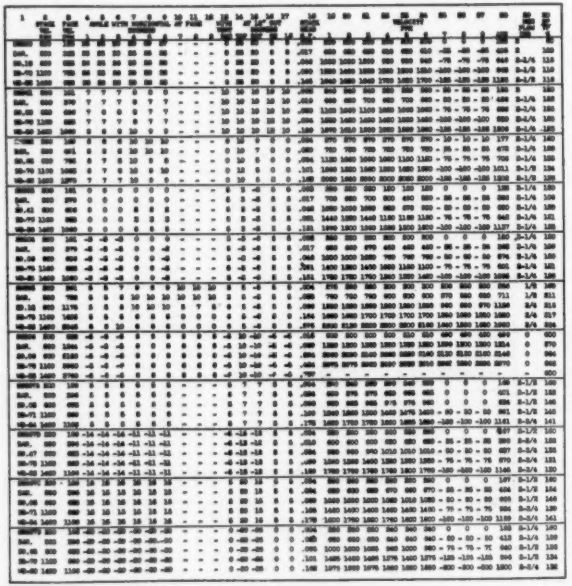
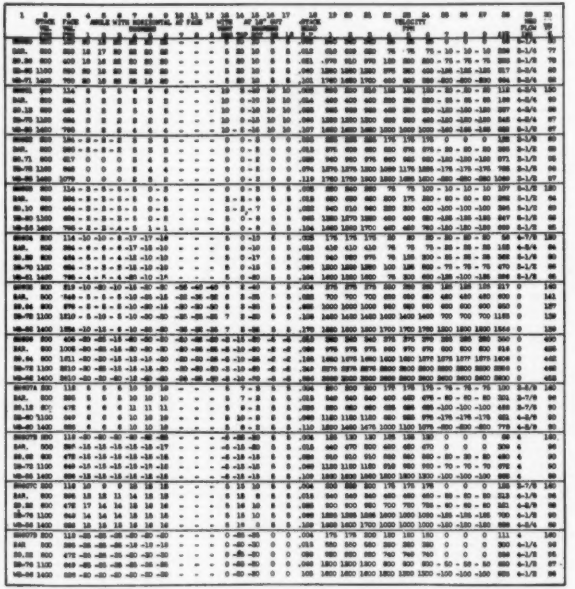


TABLE 4. RESULTS OF TESTS ON GRILLES WITH STACK HEAD No. 6



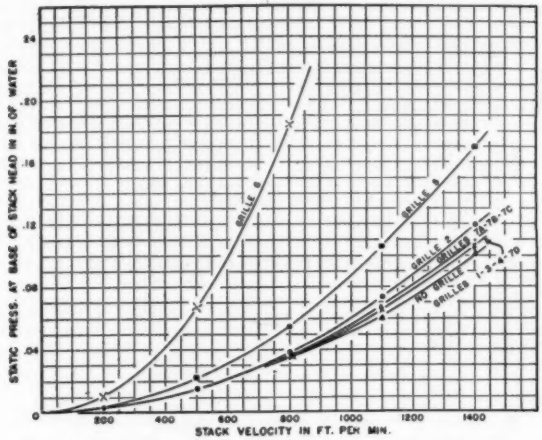


FIG. 16. VARIATION OF STATIC PRESSURE AT BASE OF STACK HEAD NO. 6 WITH VARIOUS GRILLES

ACKNOWLEDGMENT

The tests on which this paper are based extended over a period of two and a half years, with many of the tests repeated several times. Acknowledgment is especially due to the following students in mechanical engineering for their valued assistance in the program: G. E. Buske, A. C. Holmes, D. L. Hunzicker, C. L. Kaiser, W. H. Rowe, R. W. Seiders, and H. L. Thies.

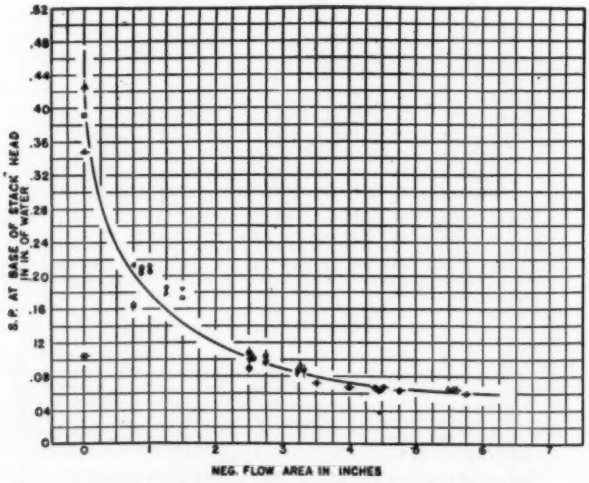


FIG. 17. STATIC PRESSURE AT BASE OF STACK HEAD PLOTTED AGAINST NEGATIVE FLOW AREA IN INCHES

No. 1207

VAPOR PRESSURE OF ICE FROM 32 to -280 F

By JOHN A. GOFF,* PHILADELPHIA, PA.

Second progress report of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Towne Scientific School, University of Pennsylvania.

INTRODUCTION

VALUES of the vapor pressure of ice for the range 0 to −90 C were published by E. W. Washburn (1)† in 1924 and later included in International Critical Tables (2). These values have since been generally accepted as sufficiently accurate except possibly for application to low temperature humidity control (3). Recently, however, important revisions in the fundamental data on which Washburn based his calculations have appeared making it desirable to recalculate the vapor pressure and extend the range to lower temperatures.

The task of recalculating the vapor pressure of ice was undertaken as a part of the cooperative investigation between the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the Towne Scientific School, University of Pennsylvania. Since this investigation has for its ultimate objective a formulation of the thermodynamic properties of moist air which, on account of accuracy and thermodynamic consistency, can claim universal acceptance as standard, and since values of the vapor pressure of ice are necessarily a part of such a formulation, it seems proper to offer the present paper as a second progress report of the investigation.

The first progress report (4), which was previously presented to the Society, gave preliminary results on the experimental measurement of the interaction constant for moist air at a single temperature of 15 C. Owing to the desirability of making some improvements in the apparatus before attempting to obtain final results covering as wide a range of temperature as possible, no additional experimental data can be presented at this time.

THE CLAUSIUS-CLAPEYRON RELATION

This identical relation of thermodynamics provides the basis for calculating vapor pressure from other known data. It is merely a mathematical statement of the conditions for co-existence of two distinct phases of a pure substance in

* Dean, Towne Scientific School, University of Pennsylvania. MEMBER of ASHVE.

† Numbers in parentheses refer to the bibliography at end of paper.

Presented at the 48th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January, 1942.

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the absence of capillary and all other external forces except pressure, as explained in Chapter 1, THE GUIDE 1942 (5). These conditions are equality between the two co-existing phases of (*a*) pressure, (*b*) temperature, (*c*) specific free enthalpy. The relation may be regarded as exact.

If h_i, v_i denote the specific enthalpy and specific volume of the saturated solid (ice) at absolute temperature T and if h_g, v_g denote specific enthalpy and specific volume of the saturated vapor at the same temperature T, then the Clausius-Clapeyron relation requires that the saturation pressure p_s vary with temperature T in the following manner,

$$\frac{dp_s}{dT} = \frac{1}{T} \cdot \frac{h_g - h_i}{v_g - v_i} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Since this relation is exact, the range and accuracy of values of vapor pressure calculated from it are limited only by the range and accuracy of available knowledge regarding the quantities appearing in its right hand member, including the absolute temperature T.

THE SPECIFIC ENTHALPY OF THE SATURATED VAPOR, h_g

As explained in Chapter 1 of THE GUIDE 1942 (5), this differs from the zero-pressure specific enthalpy h^o_w by a small amount which, at a given temperature, is proportional to the pressure; thus,

$$h_g = h^o_w - B_{ww} \cdot p_s \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

In the present problem it is to be understood that p_s denotes saturation pressure. Values of the zero-presure specific enthalpy h^o_w are obtained from

TABLE 1—RELATIVE MAGNITUDES

	1	2	3	4
t (F)	h^o_w BTU PER LB	B_{ww}/RT cm^3 per Kg	$\triangle h$ BTU PER LB	hi BTU PER LB
32	0.0	0.615	0.0452	0.0
0	− 14.199	1.032	0.0455	− 15.586
− 32	− 28.379	1.812	0.0456	− 30.171
− 64	− 42.542	...	0.0457	− 43.769
− 96	− 56.695	...	0.0458	− 56.361
−128	− 70.841	...	0.0459	− 67.962
−160	− 84.982	...	0.0460	− 78.630
−192	− 99.119	...	0.0461	− 88.363
−224	−113.251	...	0.0462	− 97.202
−256	−127.381	...	0.0463	−105.174
−288	−141.509	...	0.0465	−112.207

spectroscopic measurements. Stephenson and McMahon (6) have calculated values of the zero-pressure rotational specific heat of *normal* water vapor from 5 to 300 K. *Normal* water vapor is the 3:1 mixture of the *ortho* and *para* modifications in stable equilibrium only at temperatures above 100 K, but always observed, under ordinary conditions at all accessible temperatures below this. Their results determine the magnitude of the distortion effect due to centrifugal forces which must be added to the values of specific heat calculated by A. R. Gordon (7) for temperatures above 300 K. From these two sources can be obtained reliable values of the total zero-pressure specific heat of normal water vapor clear down to 5 K and, by integration, corresponding values of zero-pressure specific enthalpy h^o_w. Best values are listed in the column (1), Table 1, where 32 F has been taken as reference point.

Returning to Equation 2 the factor B_{ww} is a temperature function obtained by differentiating the second virial coefficient A_{ww} (5) according to the definition,

$$B_{ww} = -T^2 \frac{d(A_{ww}/T)}{dT} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Values of this quantity are obtainable from the equation of Keyes (8); and while there may be some question regarding the validity of this equation at the low temperatures considered in the present problem, it will be shown that the uncertainty introduced into the final results is very slight because the vapor pressure is so small. A few of the computed values of B_{ww}/RT are listed in volumn (2) Table 1.

THE SPECIFIC ENTHALPY OF THE SATURATED SOLID, h_i

Giauque and Stout (9) have measured the specific heat of ice at atmospheric pressure down to 15 K. By integration the corresponding values of specific

OF TERMS IN EQUATION 10

5	6	7	8	9
A_{ww}/RT cm^3 per Kg	$1218.68\left(\frac{1}{T}-\frac{1}{T_o}\right)$ BTU PER LB	$\int_{T_o}^{T} \frac{(h_{ow}-h_i)\,dT}{T^2}$	$I(p_o, T)$	$\log_{10}\left(\frac{p_o}{p_{oo}}\right)$
0.0901	0.0	0.0	0.0	0.0
0.1442	−0.172530	−0.000112	−0.000041	$\bar{1}.31968$
0.2442	−0.370877	−0.000385	−0.000056	$\bar{2}.53713$
...	−0.601304	−0.000684	−0.000059	$\bar{3}.62811$
...	−0.872280	−0.000797	−0.000059	$\bar{4}.56012$
...	−1.195539	−0.000382	−0.000059	$\bar{5}.28822$
...	−1.587829	+0.001099	−0.000059	$\bar{7}.74856$
...	−2.073906	+0.004514	−0.000059	$\bar{9}.84703$
...	−2.691967	+0.011332	−0.000059	$\bar{11}.43893$
...	−3.504216	+0.024131	−0.000059	$\bar{14}.28935$
...	−4.619225	+0.047804	−0.000059	$\bar{19}.98984$

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enthalpy are easily derived. Strictly speaking these values should be corrected to saturation pressure p_s. For this purpose the identical relation,

$$\frac{\partial h}{\partial p} = v - T\frac{\partial v}{\partial T}, \quad \ldots \quad (4)$$

may be used; and it is probably correct to assume that the righthand member is independent of pressure so that (4) may be applied in the form

$$\Delta h = \left[v - T\frac{\partial v}{\partial T}\right](P - p_s), \quad \ldots \quad (5)$$

P denoting atmospheric pressure.

Measurements of the linear coefficient of expansion β by Jakob and Erk (10), extending almost to 20 K, are available for estimating the magnitude of the bracketed term in (5). Multiplied by three, these data give the volume coefficient of expansion α defined as follows:

$$\alpha = \frac{1}{v_0}\frac{\partial v}{\partial T} \quad \ldots \quad (6)$$

where v_0 denotes the specific volume at 0 C. Thus, (5) becomes

$$\Delta h = v_0[1 + \int_0^t \alpha \cdot dT - \alpha T](P - p_s) \quad \ldots \quad (7)$$

The necessary integration is easily performed, and values of Δh are listed in column 3, Table 1. It is to be noticed that, although the absolute magnitude of Δh is considerable, nevertheless its variation with temperature, which is the only thing of importance in the final results, is entirely negligible. Consequently this correction may be ignored and the values of specific enthalpy derived from the specific heat data of Giauque and Stout (9) may be regarded as directly applicable to the saturated solid. The results are given in column 4, Table 1, the reference point being 32 F as before.

THE LATENT HEAT OF SUBLIMATION AT 32 F

In order to relate the reference-point values of h^o_w and h_1 given in Table 1, it is necessary to know the latent heat of sublimation at 32 F and, in addition, the magnitude of the term $B_{ww}p_s$ at 32 F. Osborne (11) has recalculated earlier values of the latent heat of fusion and given 333.48 ± 0.2 international joules per gram as the best value. Osborne, Stimson and Ginnings (12) have published recent determinations of the latent heat of vaporization and of the vapor pressure at 32 F and have given as best values, 2500.17 international joules per gram, 0.006228 kg per square centimeter. The magnitude of the term $B_{ww}p_s$ is readily computed from the information referred to in the section on specific enthalpy of the saturated vapor and is 0.208 Btu per pound. From these data, the best value of the difference between h^o_w and h_1 at 32 F is determined to be 1218.68 Btu per pound.

Examination of Equation 1 will show that if specific enthalpies are measured in international electrical units while pressures and specific volumes are measured in absolute units, it is important to know the relation between

the two units. The value 1.00019 absolute joule per international joule recommended by Osborne, Stimson and Ginnings (12) has been used; and this determines the conversion factor

$$1 \text{ Btu} = 778.18 \text{ ft lb} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

THE SPECIFIC VOLUME OF THE SATURATED VAPOR, v_g

As explained in THE GUIDE 1942 (5) the specific volume of the saturated vapor may be calculated from

$$p_s v_g = RT - A_{ww} \cdot p_s \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

where A_{ww} is the second virial coefficient referred to in Equation 3. A few of the computed values of A_{ww}/RT are listed in column 5, Table 1. The value of R used is

$$R = 0.110236 \text{ Btu/lb F.}$$

The specific volume of the saturated solid v_i is at most 0.000006 v_g and therefore entirely negligible so far as application of (1) is concerned.

ARRANGEMENT FOR COMPUTATION

Dividing both sides of the Clausius-Clapeyron equation by pressure p_s and incorporating the results discussed in preceding paragraphs, this equation may be rewritten as follows

$$\frac{d \log_e p_s}{dT} = \frac{1218.68 + h^o_w - h_i - B_{ww} \cdot p_s}{RT^2 \left(1 - \frac{A_{ww}}{RT} \cdot p_s\right)}$$

Rearranging somewhat and integrating, the final equation is

$$R \log_e \left(\frac{p_s}{p_o}\right) = 1218.68 \left(\frac{1}{T_o} - \frac{1}{T}\right) + \int_{T_o}^{T} \frac{(h^o_w - h_i)}{T^2} \, dT + I(p_s, T) \quad . \quad . \quad . \quad . \quad . \quad (10)$$

TABLE 2.—VALUES IN EQUATION 10 AT 8 DEG INTERVALS

t (F)	$\log_{10}\left(\frac{p_s}{p_o}\right)$	t (F)	$\log_{10}\left(\frac{p_s}{p_o}\right)$	t (F)	$\log_{10}\left(\frac{p_s}{p_o}\right)$	t (F)	$\log_{10}\left(\frac{p_s}{p_o}\right)$
32	0.0	− 48	$\bar{2}.10023$	−128	$\bar{3}.28822$	−208	$\bar{10}.71819$
24	$\bar{1}.83842$	− 56	$\bar{3}.86884$	−136	$\bar{6}.93145$	−216	$\bar{10}.09919$
16	$\bar{1}.67136$	− 64	$\bar{3}.62811$	−144	$\bar{6}.55685$	−224	$\bar{11}.43893$
8	$\bar{1}.49855$	− 72	$\bar{3}.37753$	−152	$\bar{6}.16306$	−232	$\bar{12}.73311$
0	$\bar{1}.31968$	− 80	$\bar{3}.11645$	−160	$\bar{7}.74856$	−240	$\bar{13}.97681$
− 8	$\bar{1}.13446$	− 88	$\bar{4}.84423$	−168	$\bar{7}.31167$	−248	$\bar{13}.16440$
−16	$\bar{2}.94254$	− 96	$\bar{4}.56012$	−176	$\bar{8}.85055$	−256	$\bar{14}.28935$
−24	$\bar{2}.74355$	−104	$\bar{4}.26337$	−184	$\bar{8}.36311$	−264	$\bar{15}.34406$
−32	$\bar{2}.53713$	−112	$\bar{5}.95311$	−192	$\bar{9}.84703$	−272	$\bar{16}.31966$
−40	$\bar{2}.32283$	−120	$\bar{5}.62840$	−200	$\bar{9}.29970$	−280	$\bar{17}.20571$

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where

$$I(p_s, T) = \int_{T_o}^{T} \left\{ \frac{\frac{A_{ww} p_s}{RT}(1218.68 + h^o_w - h_i) - B_{ww} p_s}{T^2\left(1 - \frac{A_{ww} p_s}{RT}\right)} \right\} dT \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

In equation 10, p_o is the vapor pressure at 32 F ($T_o = 491.70$). At present the value of p_o is only given to four significant figures (0.006228 kg per square centimeter) so that values of p itself calculated from Equation 10 are only good to four significant figures. Moreover, the calculated values of $\log_{10}(p_s/p_o)$ are subject to the uncertainty in the heat of sublimation at 32 F which is probably of the order of 2 or 3 parts in 10,000 (13); also to the uncertainty in the absolute temperature corresponding to 32 F which is probably 1 part in 10,000. Therefore the values listed in Table 3 are not to be regarded as accurate beyond the fourth significant figure with slight uncertainty in the fourth significant figure at the lower temperatures.

The solution of Equation 10 is effected in the following manner: (*a*) provisional values of p_s are calculated ignoring the term $I(p_s,T)$; (*b*) these provisional values are then used to evaluate $I(p_s,T)$; (*c*) since $I(p_s,T)$ is so very small no revision is necessary and final values of p_s/p_o are obtained directly. The relative magnitudes of the three terms in Equation 10 are shown by the figures in columns 6, 7, 8, Table 1. In this table 32-deg intervals are shown, but in the actual computations 8-deg intervals were used, the integrals being evaluated by mechanical quadrature applying the Gregory formula (14).

FINAL RESULTS

Final results are exhibited in two tables. The first, Table 2, lists values of $\log_{10}(p_s/p_o)$ at 8-deg intervals of temperature. Values at intermediate temperatures can easily be obtained by constructing a table of differences and applying Bessels' formula for interpolation to halves (14).

In Table 3 are given values of the vapor pressure of ice from 32 to − 60 F at 1-deg intervals, the unit of pressure being the inch of mercury defined as

TABLE 3—VAPOR

t (F)	$p_s \times 10$ (IN. HG)	t (F)	$p_s \times 10$ (IN. HG)	t (F)	$p_s \times 10^2$ (IN. HG)	t (F)	$p_s \times 10^2$ (IN. HG)	t (F)	$p_s \times 10^2$ (IN. HG)
...	...	29	1.5710	19	9.7904	9	5.9788	0	3.7655
...	...	28	1.4997	18	9.3280	8	5.6845	−1	3.5729
...	...	27	1.4314	17	8.8857	7	5.4034	−2	3.3894
...	...	26	1.3660	16	8.4626	6	5.1351	−3	3.2146
...	...	25	1.3033	15	8.0579	5	4.8791	−4	3.0481
...	...	24	1.2432	14	7.6708	4	4.6348	−5	2.8895
...	...	23	1.1857	13	7.3009	3	4.4017	−6	2.7385
32	1.8036	22	1.1306	12	6.9474	2	4.1794	−7	2.5948
31	1.7228	21	1.0779	11	6.6097	1	3.9675	−8	2.4581
30	1.6453	20	1.0274	10	6.2870	0	3.7655	−9	2.3280

follows: 1 in. Hg (32 F) = 0.033421 standard atmospheres = 0.034532 kg per square centimeter.

SUMMARY

The vapor pressure of ice for the range 32 to − 280 F has been calculated from most reliable information on specific enthalpies and specific volumes now available. The range could have been extended to lower temperatures since all necessary information is available down to at least − 400 F; but it was felt that the range 32 to − 280 F will supply every practical need. The data upon which the calculations are made are characterized by a high degree of accuracy and permanence; and, since the basis of calculation is one of the identical relations of thermodynamics, namely, the Clausius-Clapeyron equation, it is felt that the final results also possess a high degree of accuracy and permanence.

The final results are exhibited in the form of tables. Table 2 lists values of $\log_{10}(p_s/p_o)$, where p_o denotes the vapor pressure at 32 F, at 8-deg intervals. By constructing a corresponding table of differences these data can easily be interpolated by application of Bessels' Formula for interpolation to halves. Table 3 lists values of the vapor pressure itself for the range 32 to − 60 F at 1-deg intervals which are intended to supersede the values listed in Table 6, THE GUIDE 1942 (5).

BIBLIOGRAPHY

1. E. W. Washburn, The Vapor Pressure of Ice and Water Below the Freezing Point, *Monthly Weather Review.* Oct. 1924, p. 488.
2. International Critical Tables. McGraw-Hill Co., 1928.
3. J. H. Keenan and F. G. Keyes, Thermodynamic Properties of Steam, John Wiley & Sons, 1936, p. 14.
4. J. A. Goff and A. C. Bates, The Interaction Constant for Moist Air, ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning,* July, 1941.
5. HEATING, VENTILATING AND AIR CONDITIONING GUIDE 1942, Chapter 1.
6. C. C. Stephenson and H. O. McMahon, The Rotational Partition Function of the Water Molecule, *Journal Chemical Physics,* Vol. 7, Aug. 1939, p. 614.
7. A. R. Gordon, *Journal Chemical Physics,* Vol. 2, 1934, p. 65.
8. F. G. Keyes, Comments on the Enthalpy of Saturated Steam from 0 to 100 C and Other Steam Properties, *Journal Applied Mechanics,* Vol. 7, 1940, p. 22.

PRESSURE OF ICE

t (F)	$p_s \times 10^3$ (IN. HG)	t (F)	$p_s \times 10^3$ (IN. HG)	t (F)	$p_s \times 10^3$ (IN. HG)	t (F)	$p_s \times 10^3$ (IN. HG)	t (F)	$p_s \times 10^3$ (IN. HG)
−10	2.2042	−20	1.2591	−30	7.0081	−40	3.7929	−50	1.9923
−11	2.0866	−21	1.1888	−31	6.5993	−41	3.5613	−51	1.8649
−12	1.9747	−22	1.1223	−32	6.2125	−42	3.3428	−52	1.7451
−13	1.8684	−23	1.0592	−33	5.8468	−43	3.1367	−53	1.6324
−14	1.7673	−24	0.99929	−34	5.5011	−44	2.9426	−54	1.5265
−15	1.6713	−25	0.94254	−35	5.1743	−45	2.7596	−55	1.4270
−16	1.5801	−26	0.88879	−36	4.8655	−46	2.5872	−56	1.3335
−17	1.4935	−27	0.83788	−37	4.5738	−47	2.4248	−57	1.2457
−18	1.4113	−28	0.78966	−38	4.2984	−48	2.2718	−58	1.1633
−19	1.3332	−29	0.74401	−39	4.0383	−49	2.1278	−59	1.0860
								−60	1.0135

9. W. F. Giauque and J. W. Stout, The Entropy of Water and the Third Law of Thermodynamics. The Heat Capacity of Ice from 15 to 273 K. *Journal American Chemical Society*, Vol. 58, 1926, p. 1144.

10. M. Jakob and S. Erk, Wärmedehnung des Eises zwischen O und −253 C, *Wissentschaftliche Abhandlungen der Physikalisch—Technische Reichsanstalt*, Vol. 12, 1929, p. 301.

11. N. S. Osborne, Heat of Fusion of Ice, Research Paper RP 1260. *National Bureau of Standards*, December, 1939.

12. N. S. Osborne, H. F. Stimson, D. C. Ginnings, Measurements of Heat Capacity and Heat of Vaporization of Water in the Range 0 to 100 C. Research Paper RP 1228. *National Bureau of Standards*, August, 1939.

13. C. H. Meyers, Associate Physicist, National Bureau of Standards (private communication).

14. H. L. Rice, Theory and Practice of Interpolation, *The Nichols Press*, 1899, p. 80, 136.

DISCUSSION

L. P. HARRISON, Washington, D. C. (WRITTEN): The fundamental character of the thermodynamic relationships upon which Dean Goff has based his expression

TABLE A—COMPARISON OF THE VAPOR PRESSURE OF ICE AS GIVEN

1		2	3*a*	3*b*	4*a*	4*b*	4*c*	5*a*	5*b*
TEMPERATURE		GOFF	WASHBURN		WÄRMETABELLEN			SMITHSONIAN MET. TABLES[a]	
DEG F	DEG C	IN. HG.	IN. HG.	COL. 3*a* −COL. 2 IN. HG.	MM. HG.	IN. HG.	COL. 4*b* −COL. 2 IN. HG.	IN. HG.	COL. 5*a* −COL. 2 IN. HG.
32	0	0.18036	0.1803	−0.0001	4.579	0.1803	−0.0001	0.18032	−0.00004
23	−5	0.11857	0.1186	0.0	3.008	0.1184	−0.0002	0.11909	+0.00052
14	−10	0.076708	0.07677	+0.00006	1.946	0.07661	−0.00010	0.07733	+0.00062
+5	−15	0.048791	0.04886	+0.00007	1.238	0.04874	−0.00005	0.04931	+0.00052
−4	−20	0.030481	0.03055	+0.00007	0.772	0.03039	−0.00009	0.03084	+0.00036
−13	−25	0.018684	0.01873	+0.00005	0.471	0.01854	−0.00014	0.01890	+0.00022
−22	−30	0.011223	0.01126	+0.00004	0.280	0.01102	−0.00020	0.01133	+0.00011
−31	−35	0.0065993	0.006617	+0.000018	0.167	0.006575	−0.000024	0.00664	+0.00004
−40	−40	0.0037929	0.003801	+0.000008	0.093	0.00366	−0.00013	0.00379	0.0
−49	−45	0.0021278	0.002131	+0.000003	0.052	0.00205	−0.00008	0.00211	−0.00002
−58	−50	0.0011633	0.001163	0.0	0.029	0.00114	−0.00002	0.00114	−0.00002

[a] 32 to −49 F (0 to − 45 C) computed from Thiessen's formula. −58 F (− 50 C) computed from Nernst's formula.

EXPLANATION OF COLUMNS

Column 1. Temperature. 9 F (5 C) intervals from + 32 to — 58 F (0 to — 50 C).

Column 2. Vapor Pressure of Ice (in. Hg.) as computed by Dean Goff.

Column 3. (*a*) Vapor Pressure of Ice (in. Hg.) converted from the values given in mm. Hg. by E. W. Washburn, The Vapor Pressure of Ice and Water Below the Freezing Point, *Monthly Weather Review*, vol. 52, October, 1924.

(*b*) Differences (in. Hg.) between the values of Washburn, Column 3 (*a*), and Goff, Column 2. (*i.e.*, Column 3 (*a*) *minus* Column 2.)

Column 4. (*a*) Vapor Pressure of Ice (mm. Hg.) according to, *Landolt-Börnstein Physikalisch-Chemische Tabellen*, Berlin, 1923, vol. 2, based on Holborn, L., Scheel, K., and Henning, F., *Wärmetabellen der Physikalisch-Technischen Reichsanstalt*, Braunschweig, 1919.

(*b*) Column 4 (*a*) converted to in. Hg.

(*c*) Differences (in. Hg.) between the values in the *Wärmetabellen*, Column 4 (*b*) and Goff, Column 2. (*i.e.*, Column 4 (*b*) *minus* Column 2.)

Column 5. (*a*) The Pressure of Aqueous Vapor Over Ice, as given in *Smithsonian meteorological Tables*, Fifth Revised Edition, Washington, 1931. The values from 0 to — 49:5 C, inclusive, were computed from the formula of Thiessen, *Die Dampfspannung über Eis*. (Mitteilung aus der Physikalisch-Technischen Reichsanstalt.) Annalen der Physik, vol. 29, p. 1057, 1909. The values for temperatures below — 49.5 C from the formula found by W. Nernst in collaboration with H. Levy. (Verhandlungen der Deutschen Physikalischen Gesellschaft, Vol. 11, No. 15, p. 313, August 15, 1909.)

for the vapor pressure of ice, and the great accuracy of the basic data employed in the evaluation of the constants involved, leave little or no room for doubt as to the reliability of the vapor pressure values he has obtained within the degree of precision determined by the data in question.

The final test of the accuracy of the computed vapor pressure values is comparison of these data with carefully determined observed values or smoothed results of the vapor pressure observations. As is well known, the smoothed results are ordinarily derived with the aid of empirical equations or of equations having a partial thermodynamical ground where the constants are largely established by a least-square solution on the basis of observed vapor pressures.

The purpose of this note is to present some comparisons between Dean Goff's computed values and certain observed or smoothed values, in addition to Dr. Washburn's computed values. The results are shown in Table A and the columns are explained in the descriptive remarks following the table.

The agreement between the results of Dean Goff and of S. Weber is most remarkable, while the worst agreement is between the results of Dean Goff and those of Thiessen. The experimental determinations of Weber are considered to be more reliable than the values found by Thiessen from an empirical equation based on the

BY VARIOUS AUTHORITIES, WITH THE VALUES AS GIVEN BY AUTHOR

6		7	8a	8b	8c	8d	8e	8f
TEMPERATURE		GOFF	WEBER					COL. 8e −COL. 7 IN. HG.
DEG F	DEG C	IN. HG.	TABLE 1 DYNES/CM²	TABLE 2 DYNES/CM²	TABLE 3 DYNES/CM²	MEAN[b] DYNES/CM²	IN. HG.	
−8.95	−22.75	0.023344	791.0	790.8	790.6	790.80	0.023352	+0.000008
−12.44	−24.69	0.019273	652.9	652.8	652.5	652.73	0.019275	+0.000002
−16.92	−27.18	0.015003	508.1	508.0	508.1	508.07	0.015003	0.0
−22.70	−30.39	0.010778	365.3	365.0	...	365.15	0.010783	+0.000005
−27.49	−33.05	0.0081393	275.6	275.5	...	275.55	0.0081370	−0.0000023
−32.24	−35.69	0.0061245	207.4	207.5	...	207.45	0.0061260	+0.0000015
−40.83	−40.46	0.0035997	121.8	121.8	...	121.80	0.0035967	−0.0000030
−50.71	−45.95	0.0019010	64.25	64.25	...	64.250	0.0018973	−0.0000037
−58.99	−50.55	0.0010868	36.64	36.63	...	36.635	0.0010818	−0.0000050

[b] Mean of columns 8a, 8b, and 8c.

observed vapor pressure data of Scheel and Heuse (Ann. d. Physik, vol. 29, 1909, p. 723).

Considerable confidence in Dean Goff's results is inspired by the comparisons pre-

(b) Differences (in. Hg.) between the values in the Smithsonian Tables, Column 5 (a), and Goff, Column 2. (*i.e.*, Column 5 (a) *minus* Column 2.)

Column 6. Temperatures (F and C) at which measurements of the vapor pressure of ice were made by S. Weber, *Communications from the Physical Laboratory at the University of Leiden*, No. 150ᵃ. (Published in Danish in *Selskabets Oversigter* 1915 No. 6.)

Column 7. Vapor Pressure of Ice corresponding to the temperatures in Column 6, where the vapor pressure data were computed by use of Newton's interpolation formula from the primary values of Dean Goff (see Column 2) for each degree from 32 to — 60 F.

Column 8. Vapor Pressure of Ice (dynes/cm²) as measured by S. Weber; (see Column 6), corrected values.

(a) Vapor pressures from Weber's Table 1.[1]
(b) Vapor pressures from Weber's Table 2.[1]
(c) Vapor pressures from Weber's Table 3.[1]
(d) Mean of columns 8 (a), 8 (b), and 8 (c).
(e) Column 8 (d) converted to in. Hg.
(f) Differences (in. Hg.) between the values of Weber, Column 8 (e), and Goff, Column 7. (*i.e.*, Column 8 (e) *minus* Column 7.)

[1] The data given in Weber's Tables 1, 2, and 3, refer to different methods of measurement, and other variable experimental conditions (such as quantity of ice used, etc.).

sented, at least within the range of temperatures covered. This is exceedingly gratifying to all concerned.

WILLIAM GOODMAN, La Crosse, Wis. (WRITTEN): The basic data which Dean Goff has used as well as his mode of attack on this problem make it certain that the values he has presented for the vapor pressures of ice are highly accurate. It is interesting to note that in the range from -40 F to $+32$ F the values of vapor pressure computed from the Washburn [2] equation agree closely with the values given by Dean Goff. For example, in the range from -40 F to $+26$ F, the maximum deviation between the values computed from the Washburn equation and those given by Dean Goff is only $+0.3$ of 1 per cent with an average deviation throughout the entire range of $+0.2$ of 1 per cent. From $+27$ F to $+32$ F, the maximum deviation is less than -0.05 of 1 per cent.

The deviations for temperatures lower than -40 F were not checked. However, it is probable that the deviations will be small even for temperatures somewhat lower than -40 F because the percentage deviation decreases slowly from the maximum deviation of $+0.30$ of 1 per cent at -19 F down to $+0.23$ of 1 per cent at -40 F.

W. L. FLEISHER, New York, N. Y. I think this paper brings great credit to the Society, being one of the fundamental scientific developments in the thermodynamics of the art in which we are all engaged. I myself have felt for some time since Dean Goff agreed to carry on this work, and to credit it to our Society, that we certainly had acquired a distinction that this Society had not enjoyed before. In a way, Dean Goff's work compares with the fundamental work revolving around the development of the Keenan-Keyes Steam Tables carried on by the *American Society of Mechanical Engineers*. I really think it is the beginning of a movement to put us on a plane to which we have always aspired, but that we had not quite reached before. We owe a deep debt of gratitude to Dean Goff.

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[2] The Vapor Pressure of Ice and of Water Below the Freezing Point, by E. W. Washburn, (*Monthly Weather Review*, October, 1924).

No. 1208

PHYSIOLOGICAL FATIGUE FROM EXPOSURE TO HOT ENVIRONMENTS

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in its Research Laboratory at the Pittsburgh Experiment Station of the U. S. Bureau of Mines.

A COMMUNITY of interest has developed during the past two decades between the application of heat by the medical profession in the treatment of disease and the studies of thermal relations between man and his atmospheric environment, as carried out by the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

The early studies [1,2,3,4] of physiological reactions of persons to hot atmospheres by the ASHVE, in cooperation with the U. S. Bureau of Mines and the U. S. Public Health Service, have been accepted by the medical as well as the engineering profession. The later studies of thermal exchanges [5,6,7,8]

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[1] Some Physiological Reactions to High Temperatures and Humidities, by W. J. McConnell and F. C. Houghten. (ASHVE TRANSACTIONS, Vol. 29, 1923.)

[2] Further Study of Physiological Reactions, by W. J. McConnell, F. C. Houghten and F. M. Phillips. (ASHVE TRANSACTIONS, Vol. 29, 1923.)

[3] Air Motion—High Temperatures and Various Humidities—Reactions on Human Beings, by W. J. McConnell, F. C. Houghten and C. P. Yagloglou. (ASHVE TRANSACTIONS, Vol. 30, 1924.)

[4] Work Tests Conducted in Atmospheres of High Temperatures and Various Humidities in Still and Moving Air, by W. J. McConnell and C. P. Yagloglou. (ASHVE TRANSACTIONS, Vol. 31, 1925.)

[5] Heat and Moisture Loss from the Human Body and Its Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant. (ASHVE TRANSACTIONS, Vol. 35, 1929.)

[6] Heat and Moisture Losses from Men at Work and Their Application to Air Conditioning Problems by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant. (ASHVE TRANSACTIONS, Vol. 37, 1931.)

[7] Thermal Exchanges Between the Human Body and Its Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant. (*The American Journal of Physiology*, Vol. 88, No. 3, April 1929.)

[8] Thermal Exchanges Between the Bodies of Men Working and the Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant. (*The American Journal of Hygiene*, Vol. XIII, No. 2, March, 1931.)

Presented at the 48th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January, 1942.

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between the human body and its atmospheric enviroment as affected by the temperature, moisture content and movement of the air, and the application of air conditioning to hot industrial environment[9,10,11] by the Society were all of some little value as a background in the application of saturated conditioned air to fever therapy.[12,13]

The past several years have seen extensive developments in fever therapy and other heat treatments of disease by the medical profession which has greatly extended the knowledge of the effect of hot atmospheres, either in industry or in tropical climates, on the physiological response, comfort and well-being of man.

This paper deals with some of the clinical findings resulting from fever therapy, with an attempt to interpret their significance to the reactions of normal persons subjected to long exposure to heat, either in industry or in the tropics. Important findings in these studies have been the numerical increase in the white cell or leucocyte count of the blood following the application of fever therapy,[12,13] the relation of this increase to the clinical response, and the observed fatigue, resulting in a failure in this increase in certain cases. This experience accounts for the unsatisfactory response of normal persons exposed for long periods to high temperatures, either in hot industries or hot tropical climates, and has a very important industrial hygiene significance. A better understanding of these cases and the advantage of improving them should be obtained through the application of air conditioning.

From an analysis of 847 patients treated in a cooperative study it was possible to verify certain conclusions regarding the relationship between white count response, temperature elevation, and improvement. The vag ıe reference to *resistance, body immunity, tolerance,* etc., is not entirely clear even to the writers, but when lack of improvement fails to occur in fever treated patients, these repeated experiences have been very helpful in determining whether treatment should be continued or discontinued.

Clinically, the role of the leucocyte as it appears in so-called normal states of health from an accepted text book would be approximately 6,000 with an average of 60-70 per cent polymorphonuclear leucocytes, 25-33 per cent lymphocytes, 2-6 per cent monocytes, 1-4 per cent eosinophiles, 0.25-0.50 per cent basophiles. Disease, fever therapy, or exposure of humans to hot environments either by occupational or coincidental adaptation, may alter these figures over a wide range, and such alterations may apply to the temperature variations.

Except for such diseases as malaria, typhoid, measles, influenza, etc., it is most common to find an increase in the total white count accompanied by a rise in body temperature. This phenomenon is frequently described as *resistance to infection* or the *protective power of the human organism.* An excel-

[9] Practical Applications of Temperature, Humidity and Air Motion Data to Air Conditioning Problems, by F. C. Houghten, W. W. Teague and W. E. Miller. (ASHVE TRANSACTIONS, Vol. 33, 1927.)

[10] Air Conditioning in Industry—Physiological Reactions of Individual Workers to High Effective Temperatures, by W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten and M. B. Ferderber. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

[11] Seasonal Variation in Reactions to Hot Atmospheres, by F. C. Houghten, A. A. Rosenberg and M. B. Ferderber. (ASHVE TRANSACTIONS, Vol. 46, 1940.)

[12] Fever Therapy Induced by Conditioned Air, by F. C. Houghten, M. B. Ferderber and Carl Gutberlet. (ASHVE TRANSACTIONS, Vol. 43, 1937.)

[13] Report of the First Year of Fever Therapy Research. (*Department of Industrial Hygiene,* School of Medicine, University of Pittsburgh, 1938.)

lent example of this action was observed in pneumonia research, where those patients with leucocytosis (increased leucocytes) and high temperatures withstood the disease more satisfactorily than those having leucopenia (reduction in leucocytes) or no change along with a slight rise in body temperature. If this assumption is correct, it may be possible to expect a greater average of improvement with high white counts and fever than in any other group. This is illustrated in Fig. 1, which shows the index of improvement in proportion to the cellular increase before and after treatment.

Those diseases which have responded best to fever therapy are gonorrhea, syphilis, chorea, and certain forms of arthritis. However, under certain conditions where disease is not a factor, certain definite changes have been observed in relationships between hot environments and the temperature-white cell ratio, which will be discussed later in this report. Using identical criteria for all patients treated, the following may be postulated:

1. Failure of white cell increase indicated lack of clinical improvement.
2. Difficulty in body temperature elevation produced the same outcome.
3. The resistance to heat revealed failure in improvement in addition to lack of temperature and white cell increases.

The white cell increase was arbitrarily set at a minimum of 50 per cent of the pre-treatment figure as some indication of improvement, although a greater increase is more satisfactory. The optimum temperatures necessary for clinical improvement should fall within standardized ranges of 106-107.4 F for gonorrhea; 104-105 F for syphilis; 105 F for chorea, and 103-105 F for non-specific arthritis.

Fever therapy has proved very helpful in gonococcal infections, especially in its earlier stages. In long standing cases, temporary relief resulted where failure of leucocytic and temperature increase occurred, either after the first or subsequent treatments. Even in early cases, where fever was administered frequently at short intervals patients have become *resistant* or *immune* to heat, in addition to little or no change in white count.

During the first World War, Wagner-Yauregg inoculated malarial organisms into the blood of patients suffering from syphilis to produce fever. More recently, fever therapy has to a great extent eliminated this method for reasons of safety, control and flexibility. Here, too, the ratio of improvement to increases in temperature and white cell count previously described holds true in every respect.

Chorea, especially in children, responds quite satisfactorily to fever therapy. They too, reveal the same tendencies of white blood count, temperature relationship as the adult.

However, the arthritic presents a very interesting situation. The causes in many cases are vague, except where known infection can be demonstrated. Since controversy exists over this disease only one group shall be discussed; i.e., infectious arthritis. Usually, an early case is characterized by fever, increased white cell count, swollen, painful joints and prostration. Frequently, fever is beneficial in the majority of patients, and even though no specific measures are taken the described findings usually subside. Later, there may be a recurrence of the same illness, with smaller increases in temperature and white cell count, and treatment becomes even more difficult. These flare-ups may occur occasionally, and it has been observed that hyperpyrexia is of little

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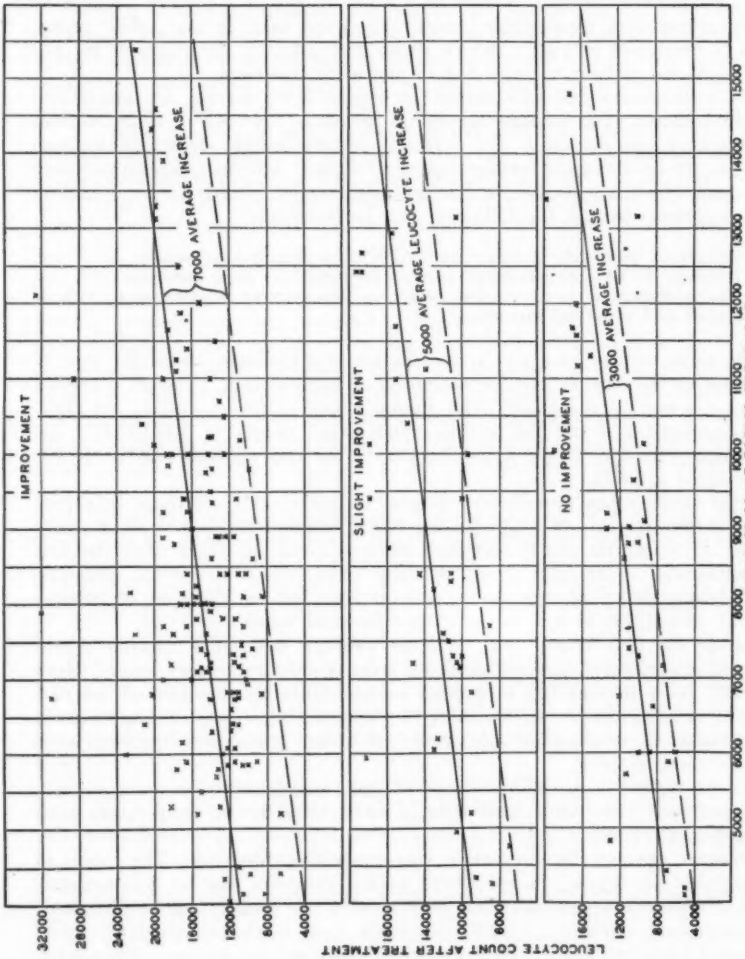

FIG. 1. RELATION BETWEEN DEGREE OF IMPROVEMENT, THE NORMAL LEUCOCYTE AND INCREASE IN LEUCOCYTE COUNT DURING 3-HOUR TREATMENT IN ABOUT 108 F SATURATED AIR

value once it has become chronic. In this stage, one finds the tender, enlarged joints without increases in fever or white count, and fever therapy apparently is of little or no value. From this, it would appear that long continued illness, however slight, would exhaust those necessary powers required for eliminating such infections. Treatment, therefore, can be directed toward the early illness before a deficiency of physiological factors occurs.

RESULTS

The experiences in these studies and those of many other investigators definitely indicate the physiological changes produced by exposures to hot environments. Since such changes may be accepted as a definite part of fever therapy, it would follow that exposures, experimental, occupational or industrial, of humans to hot atmospheres should produce some physiological changes. In 1938 the Society, with the help of the Department of Industrial Hygiene, School of Medicine, University of Pittsburgh, investigated [14] the physiological effects of hot atmospheres on human beings. These experiments were based upon the exposure of young healthy adults to environments which might be found in occupations existing in those sections of the country where high dry-bulb temperatures occur. It was observed that working in high effective temperatures of approximately 85 to 90 deg would produce increases in pulse rate, body temperature, and finally the leucocyte count. This latter finding was based on the original fever therapy work conducted [15] at the Society's laboratory, when individuals were placed in saturated psychrometric rooms. One experience is interesting, since it concerns a young Italian male subject, age 23, who showed neither temperature nor white cell increase and because of his extreme restlessness was removed from the hot chamber. He did not admit to the presence of any chronic infection until questioned, although his blood count and temperature were perfectly normal before the experiment began. Since this subject resembled patients with chronic infections previously described in this report, the first laboratory experience with individual resistance to heat was observed.

There is a normal diurnal variation in body temperature, which is usually greater in sick people; and, therefore, since the increase in white count is a function of increased body temperature, this subject must have had daily leucocytic rises. However, should this daily rise and fall continue for a very long period, it is not unreasonable to assume that a state of exhaustion or fatigue of the defense mechanism of the body would occur. This may be compared with the elementary physiology of stimulating frog muscles to a state of fatigue where no further action occurs due to overstimulation. If one grants the logic of this reasoning, it is not unlikely that long exposures to high environmental temperatures might not be conducive to good defense against infection. This may be better illustrated by experiences with pneumonia patients who have been infected during the hot summer months. In the cold months the same procedures are used, including serum, cooled oxygen tents, chemical treatment, etc.; and yet the so-called *summer* pneumonias are feared for some reason that is yet intangible. The troublesome *summer* cold

[14] Loc. Cit. Note 10.
[15] Loc. Cit. Note 12.

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is another example of failure to respond during long continued heat exposures. The same analogy may be used in any other infection; for example, gonorrhea or arthritis, where there was probably a daily rise in fever and white count, but little or no response to fever therapy previously described. On the other hand, the early case may respond satisfactorily but repeated exposures to hot atmospheres failed to produce an increase in temperature and cell count. In this case the conditions of exposure have artificially produced a status of fatigue or exhaustion of the blood cell producing (hemopoetic) and thermal regulating mechanism of the body.

From the laboratory standpoint, including the experimental industrial work, the question might be raised whether this cell count increase was due entirely to dehydration which in turn produced blood concentration. Both capillary and venous blood counts were made; and since there was little or no difference, capillary blood samples were continued to be used. The work of Krusen [16] amply illustrates the changes which do occur in the count before and after fever therapy. If dehydration is prevented by sufficient fluid intake during the treatment, there is little or no variation in hemoglobin or cell content of the blood. Differential counts on the white cells also verified the absence of dehydration. There is a relative and absolute increase of polymorphonuclear leucocytes either during or immediately after the febrile period. This numerical increase has been as high as 30,000 from a pre-treatment figure as low as 6,500.

When one considers the role of the leucocyte and its use in infection, it becomes clear that its phagocytic power (ability to destroy micro-organisms or harmful cells) is all important, either by destruction of the offending organism or removal of the products of infection. The original work of Metchnikoff [17] capably demonstrated that the polymorphonuclear leucocytes take up the bacteria into the cell, demonstrated by staining, which shows vacuolization where the organism formerly existed. Metchnikoff has also shown that phagocytosis is regularly more active in cases where animals or humans eventually recover. Such a statement would tend to bear out the investigations here reported that there is a definite relationship between the extent of improvement and phagocytosis interpreted as white cell increase.

Fever therapy and chronic infection are not the only producers of static cell and temperature responses, but high temperatures may produce the same conditions. Although individuals have an excellent thermal regulatory system, exposure to hot environments by occupation or residence produces elevated body temperatures and in turn increased white cell counts. Again, this situation is not unlike a chronic infection because the exposure is daily and similar to the diurnal temperature variation. Therefore, in cold weather, after working in hot spaces, removal to a cool environment permits a return to normal, but in very hot weather the change is either retarded or the place of residence is not sufficiently cool to produce a return to normal. The term *acclimatization* is fitting because the worker becomes accustomed to physiological changes where the working environment is not unlike the resting environment. It

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[16] The Blood Picture Before and After Fever Therapy by Physical Means, by F. H. Krusen, (*American Journal Medical Science*, 193:470, April, 1937.)

[17] Zinsser, Bayne-Jones Text Book of Bacteriology, 232-234, 1934, by E. L. Metchnikoff. (D. Appleton-Century Co.)

has been shown [18] that there is a difference in response in different seasons of the year. From workers who have spent long periods in very hot climates, there is evidence to show that the failure of physiological defense against infections is directly proportional to the time spent in the area. If true, one might conclude that the constant existence in extremely hot environments might produce a condition in the body where temperature and white cell response might fail as a defense measure.

It is frequently desirable to alleviate undesirable exposure to these hot atmospheres. Means for accomplishing this end have been established through research. Where the occupancy density in a hot industry is high, the most logical and economical is to air condition the entire enclosure so as to supply ambient air in the entire enclosure sufficiently satisfactory to give the desired results. In industrial establishments where a small number of workers are engaged in large enclosures complete air conditioning may not be feasible. Here, one of the following methods may be used as indicated in earlier laboratory studies: [19] (1) In some instances the worker may be placed in a conditioned booth from which he may observe operations and perform his required tasks. (2) The worker may be supplied with a blast of cool, high velocity air into which he may step, thus providing himself with the degree of cooling required. (3) Cooled air may be blown through a loose-fitting, relatively air-tight suit so as to cool the worker without affecting the surroundings.

DISCUSSION

W. L. FLEISHER, New York, N. Y.: My interest in investigating the results reported in this paper led me to question the effectiveness of prognosticating the rise in leucocyte count as given in a lot of the data obtained at the Laboratory where I noted a lack of rise in leucocyte count in some of the high temperature work.

Some of the data which Dr. Ferderber has obtained indicated that those people who did not have a rise in leucocyte count corresponding to an increase in body temperature were not able to withstand a good many conditions imposed on them, such as the resultant recovery from various diseases.

From similar observations made at the Society Laboratory it was found that those subjects that did not have a rise in leucocyte count were not able to withstand the test and had to be withdrawn. I feel very definitely that Dr. Ferderber is not satisfied completely with his analysis or summary, but feels it is something that should be carefully investigated.

W. R. RHOTON, Cleveland, Ohio: I like this paper because it is both theoretical and experimental, and as such the results can be applied in actual practice. Some time ago I was concerned with reference to high temperature problems and the effect on workers. In this particular application the temperatures were so high around the furnaces it scorched the clothing and in some cases even the hair. A method was devised to use a few panels consisting of copper tubing and metal lath with plaster on each side and water circulated through the tubes. These panels were installed in a rather inexpensive manner and we were able to reduce the temperature around the furnace to the extent that even a chilling effect was noted. We understood that

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[18] Loc. Cit. Note 11.

[19] Local Cooling of Workers in Hot Industry, by F. C. Houghten, M. B. Ferderber and Carl Gutberlet. (ASHVE TRANSACTIONS, Vol. 47, 1941.)

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the use of these panels even assisted in the length of life for the fire-brick located within the furnace.

L. T. AVERY, Cleveland, Ohio: I do not know whether you realize the criticism the engineer must take when he undertakes to apply air conditioning to the health of the people. I recall an incident: a physician first called me to say, "Could we provide a humidity condition to take care of asthma?" and then said, "Well, we have decided to send the patient to Arizona, because we do not know what temperature or humidity we would like you to carry." Dr. Ferderber has contributed to our Society, and we should be appreciative of this work because, while he may be a quasi-engineer, he is an air conditioning doctor.

No. 1209

PHYSIOLOGICAL INFLUENCE OF ATMOSPHERIC HUMIDITY

Second Report of the ASHVE Technical Advisory Committee on Physiological Reactions: C.-E. A. Winslow, Dr. P.H., *Chairman;* Thomas Bedford, Thomas Chester, E. F. DuBois, M.D., M. B. Ferderber, M.D., E. P. Heckel, John Howatt, R. W. Keeton, M.D., C. S. Leopold, R. R. Sayers, M.D., Charles Sheard, Ph.D., and C. Tasker.

THREE years ago this Committee in its first report summarized certain general principles revealed by physiological research with regard to the desirable objectives of air conditioning. A considerable amount of new knowledge has been accumulated since 1939, particularly in connection with the influence of atmospheric humidity. In the present report, it is proposed to review the present state of information in this field with respect to the three essentially different problems involved: the influence of atmospheric humidity in cool air upon heat losses from the human body; the influence of atmospheric humidity in warm air upon heat losses from the human body; and the influence of atmospheric humidity upon the dryness of the membranes of the nose and throat.

Influence of Atmospheric Humidity on Human Body

In Cool Air Upon Heat Losses

Since one of the important avenues of heat loss from the human body is evaporation, and since a moist atmosphere tends to reduce evaporative heat loss, high relative humidity should be theoretically desirable in a cool environment. Quantitatively, however, this protective influence of a moist atmosphere is relatively unimportant for two reasons.

In the first place (as was pointed out in the first report), the extent of evaporative heat loss is small under such conditions. At 68 F, for instance, the heat loss from the clothed human body due to radiation plus convection is three times that due to evaporation (even in dry air). In the second place, the difference between the vapor pressure of a cool atmosphere (which determines evaporative heat loss) under moist and dry conditions is relatively small, as compared with the vapor pressure at the surface of the hot skin.

At 70 F, for example, the vapor pressure of water in air 75 per cent saturated is 0.56 in. of mercury and in air 25 per cent saturated is 0.18 in. of

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mercury. The temperature of the body surface is about 90 F and the corresponding vapor pressure 1.42 in. of mercury. The effect upon sweat evaporation of changing relative humidity from 25 per cent to 75 per cent will be proportional to the difference between 1.42-0.18, and 1.42-0.56, or between 1.24 and 0.86, a reduction of about 30 per cent. At 70 F, even in dry air, the loss of heat by evaporation is only 20 to 30 kilogram-calories per hour, hence change to a moist atmosphere can reduce the evaporative heat loss by a maximum of about 10 kilogram-calories.

This conclusion is justified by ample experimental evidence. It has been shown for the nude body at rest (Gagge, Herrington, and Winslow, 1937) for the clothed body at rest (Winslow, Herrington, and Gagge, 1938), and for the nude body in active work (Winslow and Gagge, 1941) that evaporative heat loss in cool air is less than 10 kilogram-calories greater in dry air (25 per cent relative humidity) than in moist air (75 per cent relative humidity). This increased heat loss in dry air is equivalent to a decrease in air and wall temperature of 2 F for the nude subject and 2.5 F for the clothed subject (Gagge, Winslow, and Herrington, 1938).[1]

These experimental results on heat loss, therefore, indicate that at low air temperatures it takes an increase of 20 to 25 per cent in relative humidity to compensate for a decrease of 1 F in the air temperature. This corresponds closely to a value of 23 per cent derived from the line on the ASHVE Comfort Chart for an air temperature of 60 F.[2] It is also checked by the work of Bedford (1936 and 1939) in England which indicated 24 per cent as the change in relative humidity (in cold air) equivalent to 1 F change in temperature, so far as the sensation of warmth is concerned. Bedford's more recent work (Bedford and Warner, 1939), furthermore, shows that the sensation of *freshness,* as distinct from that of *warmth* (at moderate temperatures) is *inversely* correlated with relative humidity. It is probable that this latter phenomenon is related to sensations aroused in the nose and throat, or it may possibly have some relation to odors. It seems clear, then, that at comfortable and cool temperatures humid air is psychologically undesirable and that its thermal influence in reducing evaporative cooling is negligible. There is, therefore, no physiological basis for humidifying air at moderate or low temperatures on thermal grounds.

The temperature zone in which the influence of atmospheric humidity is relatively slight will, however, vary with the metabolism of the subject, the clothing which is worn and the rate of air movement. It is limited to those conditions under which the body is not actively sweating; and, as will be pointed out in a succeeding paragraph, this critical point may vary from 60 F air temperature for a clothed subject doing active work in still air, to 85 F for the unclothed subject at rest.

As Yaglou and Messer (1941) have pointed out, certain factors such as dress are of greater practical importance in determining thermal comfort in the cool zone than a small humidity effect. They report that women in their usual thin clothing are most comfortable at 76 F, men in their warmer clothing at 71.5 F.

Although there have been relatively few studies on the responses of women

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[1] Throughout this report (except where otherwise stated) computations are based on such a minimum air velocity (20 fpm) as is found under ordinary indoor conditions.

[2] At 65 F the corresponding figure is 16 per cent; at 70 F, 11 per cent—all for clothed subjects.

to changes in the environmental temperature, such studies have revealed certain interesting sex differences. Hardy, Milhorat, and DuBois (1941) reported results on eight women studied nude in a respiration calorimeter. The heat production of the women in the cold zone was approximately the same as that of men, but the heat loss paralleled heat production much more closely than in the case of men. They also had a skin temperature about 1.8 F lower than that of men in the cold zone. This was apparently the result of better heat insulation against cold.

Under very cold conditions, high atmospheric humidity may be definitely harmful. It is well known that outdoor air is more chilling on a cold, damp day than on a cold, dry day. This phenomenon has not been studied in the laboratory. It is obviously related, not to any direct influence of atmospheric humidity upon evaporative heat loss but to increased conductive power of clothing which has absorbed moisture and consequent increase in heat loss by radiation and convection. This phenomenon can only occur under such conditions of fog and rain as will super-saturate the clothing.

In Warm Air Upon Heat Losses

In a warm atmosphere, where evaporative cooling plays a major role in the thermal adjustment of the body, the whole picture changes and relative humidity becomes a major factor in comfort.

The actual air temperature at which this occurs will vary widely according to various factors—of which clothing and metabolic activity are among the most important. The nude resting subject (semi-reclining in a chair) does not begin active sweat secretion until the air temperature reaches 85 F (Winslow, Herrington, and Gagge, 1937), while for the clothed resting subject, the limit is at about 75 F (Winslow, Herrington, and Gagge, 1938). For the unclothed subject doing active physical work (riding a stationary bicycle) the onset of active sweat secretion begins at an air temperature below 68 F (Winslow and Gagge, 1941). For the clothed subject doing active work, the limit would, no doubt, be considerably lower, perhaps under 60 F.

The limits for women will tend to vary moderately from these values as a result of their somewhat individual response to temperature changes. Hardy and Milhorat (1939) reported that three normal women showed a considerable drop in heat production in a warm environment. This effect was later confirmed by the study of Hardy, Milhorat, and DuBois (1941) in which they noted that the decrease in the heat production of women in the warm comfort zone ranged from 9 to 21 per cent. As a result of this lowered heat production they did not need to sweat so much in the warm environment.

At any point above the general limits mentioned previously, evaporative cooling is a vital factor in comfort, and atmospheric humidity becomes of primary importance.

In this Zone of Evaporative Regulation, the human body shows a remarkable power of adaptation. As atmospheric (and radiational) temperatures rise above the critical point, the body responds by a beautifully balanced increase in sweat secretion so that evaporative cooling keeps pace with the need for heat dissipation. For example, one subject working on a stationary bicycle at a metabolic rate of about 300 kilogram-calories per hour was losing 250 kilogram-calories per hour by convection plus radiation and less than 50 by evaporation at a temperature of 56 F. At a temperature of 91 F the

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subject was losing less than 60 kilogram-calories by convection plus radiation and 240 kilogram-calories by evaporation.

While thermally effective, this process is accompanied by definite sensations of discomfort. The first condition just described for the working subject (56 F) was described by the subject as *cool* and neither particularly pleasant nor particularly unpleasant. The second condition (91 F) was reported as *very hot* and *very uncomfortable*. Yet thermal balance was successfully main-

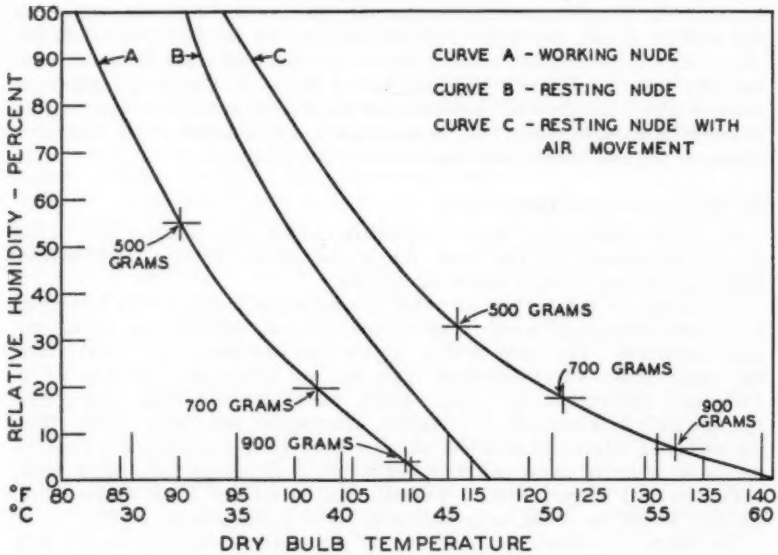

FIG. 1. UPPER LIMITS OF EVAPORATIVE REGULATION

Curve A, working, heat production 425 cal per hour, air movement 15 fpm. *Curve* B, resting, heat production 85 cal per hour, air movement 15 fpm. *Curve* C, resting, heat production 85 cal per hour, air movement 100 fpm. Crosses indicate points in limits which can be reached with indicated water loss in grams per hour.

tained in both instances. Unpleasant sensations are reported at a short range above the temperature where active sweat secretion sets in (91 F and 75 per cent relative humidity or 99 F and 25 per cent relative humidity for the nude resting subject, 64 F with 40-50 per cent relative humidity for the nude working subject). In this range, the control of relative humidity is a primary problem of air conditioning, either decrease of temperature, decrease of humidity or increase of air movement having a powerful effect on the promotion of human comfort.

Above a certain limit, this process of evaporative heat regulation becomes impossible. This limit is set by one of two different conditions. In very hot and very dry air, it is fixed by exhaustion of the power of the body to

produce enough moisture to be evaporated. In air containing much moisture, on the other hand, the limit is set by the failure of sweat to evaporate from the body surface and of course, this is the more usual condition.

Obviously, adaptive evaporative cooling can never take place in completely saturated air at or above body temperature; and in a hot atmosphere—where evaporative heat loss is vitally important—the relative humidity can be predicted for any given air temperature which will enable the body to dissipate by evaporation the amount of heat which will keep it in thermal balance. This limit can be computed (Winslow, Herrington, and Gagge, 1937) for any given rate of metabolism by constants derived in the John B. Pierce Laboratory for skin temperature, rate of convection loss and sweat secretion. Constants for these factors derived from recent studies have been used by L. P. Herrington to plot graphs, one of which is reproduced in Fig. 1. It indicates the upper limits of evaporative regulation for unclothed subjects (A) working at a metabolic rate of 425 kilogram-calories per hour in relatively still air, (B) at rest in relatively still air, and (C) at rest in air moving at a rate of 100 fpm.

At a temperature of 80 F both resting and working subjects can adjust in still air even at 100 per cent relative humidity. The resting subject can also adjust with any relative humidity at 90 F and the resting subject in rapidly moving air (100 fpm) can adjust with any humidity up to 93 F. The general slope of the lines below these points indicates the limits of possible adjustment based on the absorptive capacity of the atmosphere. The resting nude can, of course, adapt to much more extreme conditions than the working subject, and the zone of adaptation is further broadened by a high degree of air movement (100 fpm).

In line A of Fig. 1, it will be noted that at 90 F the upper limit for the working subject is at 55 per cent relative humidity; if this line were continued downward and to the right on a basis of atmospheric absorptive power, the working subject would adjust at 102 F and 20 per cent relative humidity, and at 110 F and 3 per cent relative humidity. At these points, however, the other limiting factor, the power of the body to produce sweat, comes into the picture. In this respect individuals vary widely; and the same individual may greatly increase his ability to secrete sweat. Dill (1938) has shown that a subject working in hot, dry, desert air, can double his capacity of moisture production, by practice, from 0.6 kg to 1.2 kg per hour (from a little over a pint of water to a little more than a quart). In football games much higher values have been recorded. If, however, we assume a representative figure for the normal individual, approximating Dill's initial value, it is found that above 90 F the working subject would not be able to produce the amount of moisture necessary for adjustment, even though the atmosphere could absorb it. For the working subject producing 700 g (grams) of moisture, this limit of secreting power would come at 102 F, and for the subject secreting 900 g at 110 F. Vertical lines drawn downward on the graph would indicate these limits, and similar limits are set for the resting nude subject in moving air; for the subject producing 500 g at 114 F, for the subject producing 900 g at 123 F, for the subject producing 900 g at about 132 F. At any point below the slope of the graph and to the left of the appropriate vertical line the subject can maintain thermal balance. Above the slope and to the right of the vertical line, the subject cannot.

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The determination of these upper limits is not of concern to the ventilating engineer in ordinary practice; but it is clearly vital in connection with certain industrial processes and with the air conditioning of naval vessels.

Upon Dryness of Membranes of Nose and Throat

Quite independent of such influences of atmospheric humidity upon thermal interchanges as have been discussed previously, is the direct influence of a dry atmosphere upon the mucous membranes of the nose and throat. Huntington (1919, 1924) has presented convincing evidence that dry climates and seasons are associated with definite increases in mortality rates; but the nature of the influence exerted has been little understood. Many physiologists have shown that, in general, the expired air approaches body temperature and 100 per cent relative humidity. Therefore, it is clear that the moisture taken from the membranes of nose and throat in respiration must vary with the temperature and humidity of the inspired air. The most exhaustive study of this subject was presented to the Society by Seeley in 1940. Seeley found that (except in very cold atmospheres) the expired air had a temperature of 90-95 F and was over 90 per cent saturated with moisture, containing 30-37 g of water per cubic meter of respired air. Since this value is approximately constant, it is clear that the drying effect on the mucous membrane must be related to the absolute—not the relative humidity of the atmosphere.

Recent studies at the John B. Pierce Laboratory of Hygiene (Winslow, Herrington, and Nelbach, 1941) have directly confirmed this conclusion. In these experiments, direct observations were made of the moisture actually present on the surface of the back of the throat (as measured by the amount of moisture absorbed by blotting paper from a given surface under standard conditions). The relation of the phenomenon to absolute humidity was confirmed; and an interesting critical point in the process was determined.

At atmospheric water vapor pressure below 0.4 in. of mercury at any temperature (from 50 to 80 F) the throat surface was very definitely dry. Between 0.4 and 0.5 in. a very marked increase in moisture was manifest, equal to a sudden doubling of the water absorbed by the blotting paper. At a still higher vapor pressure a secondary decrease in moisture on the mucous surface was observed at certain points (70 F with 70 per cent relative humidity, and 80 F with 50 per cent relative humidity) which is believed to be associated with vasomotor reactions related to the onset of sweat secretion. The point of primary interest, however, is the critical point at 0.4 in. vapor pressure. This finding means that marked drying of the membranes of the nose and throat must occur at air temperatures below 53 F with any moisture content up to saturation; at a dry-bulb temperature of 60 F with less than 77 per cent relative humidity; at a dry-bulb temperature of 70 F with less than 54 per cent relative humidity; and at a dry-bulb temperature of 80 F with less than 39 per cent relative humidity.

It should be emphasized that there is no direct evidence that the drying of the mucous surfaces is actually undesirable. The general opinion that very dry air irritates the throat and the clinical practice of treating the membranes of persons suffering from bronchial disease by soothing inhalations would seem to suggest that the problem may be of importance. If dry air is harmful, it seems obvious that its influence must be present, not merely in hot, dry air, as has often been thought, but quite as much in cold, dry air.

Furthermore, it appears that, if it is desired to control the influence of dry air on the mucous membranes, a relatively high vapor pressure must be maintained corresponding at 70 F to over 50 per cent of saturation.

BIBLIOGRAPHY

Recent Advances in Physiological Knowledge and Their Bearing on Ventilation Practice. Report of the Research Technical Advisory Committee on Physiological Reactions, C.-E. A. Winslow, Chairman. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

Bedford, Thomas, 1936. The Warmth Factor in Comfort at Work. (*Industrial Health Research Board* (London), Report No. 76.)

Bedford, Thomas, 1939. (*Journal Institution Heating and Ventilating Engineers* (London), VII, 377.)

Bedford, Thomas, and Warner, C. G., 1939. (*Journal of Hygiene*, 39, 498-511.)

Dill, D. B., 1938. Life, Heat and Altitude. (*Harvard University Press.*)

Gagge, A. P., Herrington, L. P., and Winslow, C.-E. A., 1937. (*American Journal Hygiene*, 26, 84-102.)

Gagge, A. P., Winslow, C.-E. A., and Herrington, L. P., 1938. (*American Journal Physiology*, 124, 30-50.)

Hardy, J. D., and Milhorat, A. T., 1939. (*Proceedings Society for Experimental Biology and Medicine*, 41:94-98.)

Hardy, J. D., Milhorat, A. T., and DuBois, E. F., 1941. (*Journal of Nutrition*, 21:383.)

Huntington, E., 1924. World Power and Evolution. (*Yale University Press*, New Haven.)

Seeley, L. E., 1940. Study of Changes in the Temperature and Water Vapor Content of Respired Air in the Nasal Cavity. (ASHVE TRANSACTIONS, Vol. 46.)

Winslow, C.-E. A., and Gagge, A. P., 1941. (*American Journal Physiology, 134*, 664-682.)

Winslow, C.-E. A., Herrington, L. P., and Gagge, A. P., 1937. (*American Journal Physiology*, 120, 258-299.)

Winslow, C.-E. A., Herrington, L. P., and Gagge, A. P., 1938. (*American Journal Physiology*, 124, 692-703.)

Winslow, C.-E. A., Herrington, L. P., and Nelbach, J. H., 1941. (*American Journal Hygiene.*)

DISCUSSION

W. L. FLEISHER, New York, N. Y. (WRITTEN): This paper revives once more the moot question of the necessity for water vapor in the atmosphere, particularly in winter. One would gather from the paper that from the standpoint of thermal equilibrium the relative humidity of the air, in winter particularly, plays a very unimportant part. I feel that we are placing too much emphasis on thermal equilibrium as measured by rather crude devices or instruments. The sensation of dryness is distinctly unpleasant and in most instances calls for an increase in sensible temperature, which accentuates, due to the lower relative humidity, the uncomfortable feeling that accompanies too dry an atmosphere.

I have a very strong feeling that a certain amount of equilibrium is established possibly by electrical emanations between the body and the surrounding surfaces, and where the surfaces themselves are fairly dry and the surrounding atmosphere is dry, or we may say below a certain absolute humidity, the body is giving off certain fundamental electrical emanations which, under higher relative humidities, would not be given off. Just as we find in industrial applications that static electricity is eliminated at a fairly high relative humidity—say 50 per cent or over—so the body, possibly from the extremities (the fingers or other uncovered parts of the body, particularly sharp points), is disturbing essential sensory equilibria which would not occur at relative humidities above the points I have mentioned.

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In a discussion of the effect of relative humidity on the feeling of comfort or the reactions of human beings to their environment with Professor Missenard in Paris several years ago, he claimed that our assumptions in connection with the effect of relative humidity had never been properly interpreted because we had never gone to the particular relative humidities which indicated change in reactions. He talked to me of 65 per cent relative humidity, which is a condition almost impossible to obtain in the wintertime in our cold climates, due to condensation on walls and windows. I am bringing this matter up simply because I do not believe that this subject can be dismissed without carrying the investigation to the optimum point, even though that particular point might not be practical.

From the angle of discussion of the Seeley paper,[3] I feel so definitely that this paper was an original contribution of tremendous importance, that it cannot be dismissed in a paragraph and that other features of this paper must be introduced into any discussion of this kind. In a discussion which I have had with Professor Seeley, we felt that the work that had to be done by the mucous membranes of the nose and throat was a function of the change in total heat between the introduced air and the resultant enthalpy of the air taken into the lungs. In other words, air saturated at 98 deg and 30 per cent relative humidity would have an enthalpy of 21.95 Btu per pound. Therefore, for each pound of air introduced, 45.97 Btu would have to be provided by the very small area of mucous membrane in the nasal passages. This might mean 20 per cent of the total thermal equilibrium that had to be made up by the whole surface of the body. Consequently, the relationship of percentages as set up by this paper must be oriented to the percentage of surface which has to accomplish the desired results. This particular amount of work, oriented against the areas involved, I think has not been properly appreciated. I therefore agree that this requirement for equilibrium is as much a function of cold, dry air as it is of hot, dry air.

There is one other point that I think has to be brought out in connection with this paper, a matter that I have discussed since presentation of a paper by me in 1931 on the effect of the pH concentration on the retention of moisture in a colloidal material. In this paper I brought out the fact that there were two kinds of moisture in any colloidal material, one being moisture of composition and the second being moisture of addition. The moisture of composition has an entirely different character than the moisture of addition. The moisture of addition is easily dissipated under normal temperatures. The moisture of composition, which is held tenaciously on the colloidal surfaces, can only be removed or dissipated under much greater bombardment, either electrical or thermal. In dealing with subjects under high dry-bulb temperatures, where the thermal equilibrium is entirely maintained by evaporation, a point is eventually reached where the moisture of addition is entirely or almost entirely dissipated. At this point it cannot be expected that the body can evaporate any moisture to maintain its thermal equilibrium, and this is probably the reason why men subjected to very high temperatures will function for a certain length of time without undue stress and then will succumb to coma.

It is therefore my contention that in all tests made at high temperatures and low humidities, unless there is definite regulation of the amount of liquid imbibed by the subject during the test and exactly the same amounts allotted to each subject, no definite conclusions can be drawn.

I suggest as a corollary to this report that this whole subject of the colloidal nature of the human body and its change in isoelectric state be determined by the constant observation of the moisture of addition factor as a determining point in the reaction of human beings to unfavorable environments.

G. DONALD FIFE, Washington, D. C. (WRITTEN): It seems that one of the conclusions to be drawn from this paper is that the physiological desirability of winter

[3] Study of Changes in the Temperature and Water Vapor Content of Respired Air in the Nasal Cavity, by Lauren E. Seeley, (ASHVE TRANSACTIONS, Vol. 46, 1940.)

humidification is open to question. The statement that "There is, therefore, no physiological basis for humidifying air at moderate or low temperatures on thermal grounds" is not surprising, in view of the researches relative to effective temperatures. However, together with the section which treats of the influence of humidity upon the dryness of the nose and throat membranes, a case is presented for eliminating the use of humidifying equipment during the winter season. The humidities which can be maintained in the winter time are limited by the necessity of avoiding excessive condensation on the windows, and this paper shows that such limited humidities fall far short of the requirements necessary to avoid drying of the nose and throat membranes. The evidence offered is consistent with the statements in the chapter on Physiological Principles in THE GUIDE. On the other hand, the paper presented by Professor Seeley at the June, 1940, meeting suggests that humidification is beneficial to health as it reduces the sudden alteration in moisture loss from the nasal cavity when one goes indoors on a cold winter's day.

I believe that this question of the advantages of winter time humidification is very important and that the uncertainties arising from conflicting statements should be cleared up if possible. The prevalent belief, and one that has been sold to the public for a long time, is that indoor air should be humidified in the winter time; not only because it benefits the inanimate objects in the conditioned spaces—such as furniture and books—but because it is also beneficial to the health of the occupants. If it is considered that there is sufficient evidence available to refute this belief, it should receive the proper publicity. Perhaps there is a lot of steam for preheaters and power for spray pumps being wasted at the present time. Or, on the other hand, if the evidence is not sufficiently conclusive, it would seem that further investigations would be valuable.

C. J. STERMER, Chicago, Ill. (WRITTEN): This paper prompts a question as to whether some of the respiratory troubles attributed to highly heated dry rooms are due to the indoor condition, or the outdoor condition during the winter season.

Data here presented indicate rather conclusively that the absolute humidity might be the important factor to investigate from a health standpoint, rather than relative humidity.

In this connection it might be helpful to study the respiratory ailments if any occurring in cold storage plant workers, who spend considerable time during the summer season in a low atmospheric water vapor pressure environment and then immediately change to a high water vapor pressure and temperature environment when passing from the cold storage space to the outdoor summer air, as compared to a winter condition when these same individuals, as well as the general public in the northerly latitudes, live continuously in an environment of low water vapor pressure though changing from low to high temperature conditions.

R. R. SAYERS, Washington, D. C. (WRITTEN): In connection with the data presented in this report it may be of interest to recall some of the results obtained by Bureau of Mines investigations in hot mines, especially with regard to the effects of exposure to hot, still, saturated air and hot, saturated moving air. Persons remaining at rest in saturated, still air at 91½ F for one hour experienced an increase in body temperature; a moderate increase in pulse rate; profuse sweating; and after effects of dizziness and weakness. In the same air conditions with air movement they experienced slight or no increase in body temperature; slight increase in pulse rate; slight perspiration; no after effects; and no ill effects at any time. Persons remaining at rest in saturated, still air at 95 F for one hour experienced an increase in body temperature; a marked increase in pulse rate; very profuse sweating, clothing being saturated with perspiration and sweat in shoes of all subjects; dizziness on movement, and increase in depth and rate of respiration (puffing somewhat on slight movement); chilly sensations in some subjects. Exposure to the same conditions with air movement caused slight or no rise in body temperature or pulse rate; profuse sweating, but not enough to wet all clothing; no untoward symptoms other than profuse sweating. Persons remaining at rest in saturated, still air at 100 F experienced a

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marked rise in body temperature, which reached 102.3 F; a marked rise in pulse rate, varying in different subjects from 152 to more than 175; profuse sweating, the shoes being partly filled with perspiration; and early appearance of dizziness, weakness, and persistence of symptoms for about one hour after test. The test was very trying. With air movement all the given symptoms were experienced and no subject remained a full hour in this atmosphere.

It is interesting to note that in one of our studies it was found that length of exposure to different high temperatures is a determining factor in the severity of the after depression produced in a subject. Although the physiological reactions may deviate further from the normal in higher temperatures for a short period of time than in a temperature not so high for a longer period, the fatigue resulting from the latter test is more prolonged than in the former. In an experiment in which the subjects remained for one and a half hours in the test chamber and the rectal temperature rose to 101.4 F and the pulse rate to 149 pulsations per minute, the fatigue and weakness which followed lasted for a longer period than the fatigue and weakness which followed in another experiment in which the rectal temperature rose to 103 F and the pulse to 162 with exposure for a period of only 45 min.

With regard to the effect of the dry and humid air on the mucous membranes, although the addition of humidity to the air in winter in air-conditioned buildings prevents this condition it apparently has no appreciable effect on the feeling of irritation in the throat which causes coughing in those suffering with colds. The drying effect of the warm air in the unconditioned, heated buildings is noticeable to those who return home to such an atmosphere after spending the day in an air-conditioned building where moisture has been added to the atmosphere. Probably most people have observed the drying effect on the mucous membranes of cold, dry air, especially with strenuous exercise in such air. With reference to the clinical practice of treating the membranes of persons suffering from bronchial diseases by soothing inhalations, it has been reported lately in some instances that medicine prescribed for coughs incident to colds has a considerable drying action on the mucous membranes of the mouth and throat.

DR. WINSLOW: I am glad that Mr. Fife called attention to the importance of this problem. If our observations as to the critical point of 0.4 inches of mercury are correct, it means that effective humidification in cold climates cannot be obtained. For the present I do not believe the results should be publicized, for I think the data should be checked by other investigators. The data on which these conclusions are based are presented in the last issue of the *American Journal of Hygiene*.

I was greatly interested in Mr. Fleisher's contribution. It is a little outside my field, but I would gladly accept him as an authority on that subject, because if anybody gave off electric vibrations from his fingers and every other part of his person, it is Walter Fleisher.

No. 1210

SANITARY VENTILATION †

By W. F. Wells,* Philadelphia, Pa.

PRIOR to the streamlined air conditioned era, heating for comfort and ventilating for health were accepted notions. Ventilation of heated confined spaces, vitiated by occupancy, was regarded as a health essential. Pettenkofer, the father of hygiene, proposed in 1858 the familiar carbon dioxide method for measuring the atmospheric density of occupants. Dilution of exhaled carbon dioxide with 20 cu ft of air per person per minute yields approximately one part per thousand. DeChaumont (1875) perceived an unpleasant odor in confined spaces when exhaled carbon dioxide exceeded six parts per ten thousand, for many years accepted as a limit by sanitarians.

Noxious vapors and exhaled effluvia are no longer considered infectious. Though bacteriology and the germ theory of disease were born together with Pasteur's classic experiments on bacteria in habited atmospheres, perfection of bacteriologic techniques during the last two decades of the 19th century identified specific agents of disease, and disclosed in rapid succession their transmission by other routes than air. Failures to recover nasopharyngeal organisms on plates exposed beyond an arm's length of a coughing or sneezing person convinced Flügge that infected droplets settled out in short distances and in brief time intervals.

Sanitary bacteriology during the next 20 years demonstrated general bacterial contamination, but successful sanitary control of infections ingested in water, milk or food, or conveyed by insects, was not extended to respiratory infections. These *catching* diseases might perhaps spread without vehicles, for obviously direct transfer by personal contact was possible among aggregations within enclosed atmospheres. Discarding Pettenkofer's teachings, the ventilation engineer pronounced fit to breathe air which satisfied comfort requirements, and turned to factors of humidity, temperature and air motion to explain the physiological effects of vitiated air. Environmental control techniques soon met these new specifications of ventilation, now called air conditioning.

New evidence gathered during the last 10 years, with new instruments and techniques, has reopened the case against air as a vehicle of infection. Better

† From the Laboratories for the Study of Air-borne Infection, supported by a grant from the Commonwealth Fund.

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understanding of state of air-suspended infection, the role of droplet evaporation, and the behavior of viruses in respiratory contagion, dispelled many cherished fallacies. The effect of ventilating factors upon the dispersion and viability of organisms in passing from person to person in an enclosed atmosphere was studied by suspending test organisms at one point and sampling at another. Improved methods of disinfection eliminated microorganisms from semi-enclosed atmospheres at 10 to 100 times the rate practically attainable by mechanical means of ventilation. Since disinfection rate depends upon the type of organism and the state of suspension, definition by comparison

TABLE 1—SANITARY VENTILATION

	GERMANTOWN	SWARTHMORE	
FIXTURE	CENTRAL	WALL	
SOURCE	SUSPENDED INDIRECT	BEAM ELEVATION	
		60 Deg	25 Deg
Volume (cu ft)	6,000	7,000	7,000
Total lethal flux (mw)	2,000	20,000	20,000
Uniform cubic flux density (mw/ft²)	6.06	54.7	54.7
Average flux density (mw/ft²)	1.26	7.64	25.5
Lethes per minute	0.8	2.0	20.0
FACTORS			
Irradiation factor	0.21	0.14	0.47
Exposure factor	0.63	0.26	0.78
Sanitary Ventilation factor	0.13	0.04	0.37
SANITARY EQUIVALENT PURE AIR REPLACEMENT (CFM/OCCUPANTS)			
Foot-lethes per occupant per minute	200	500	5,000

with removal rate by air replacement strictly applies only to a particular organism. Elimination of a standard suspension of a standard organism, equivalent to the removal by one air change, defines the lethe.

The peculiar vulnerability of air-suspended microorganisms to photobiotic effect of ultraviolet radiation offered practical means to test the hypothesis that the semi-enclosed atmospheres of our habitations in cold weather provide the vehicle for the epidemic spread of contagion. During last winter the largest epidemic of measles in the history of Philadelphia provided unusually significant epidemiologic data on the protection afforded by radiant disinfection in three primary schools. An intensive study of sanitary ventilation in the schools was undertaken to determine the relationship between sanitary and hygienic performance, between dosage and prevention. Interpretation of radiation into sanitary ventilation defined by these studies is outlined in Table 1 and Fig. 1.

Total lethal flux (in milliwatts) represents lamp ratings given by the manufacturer. Uniform cubic flux density represents the maximum flux density (in milliwatts per square foot) obtainable in a cube of the given volume by the given flux. It is obtained by dividing the total lethal flux by the cube root of the volume squared. Generation of lethal flux is an electrical problem.

Average flux density is obtained by dividing milliwatt feet of actual irradiation by the volume. Radiant flux (in milliwatts) multiplied by distance (feet) through which that flux acts gives irradiation in milliwatt feet. The same

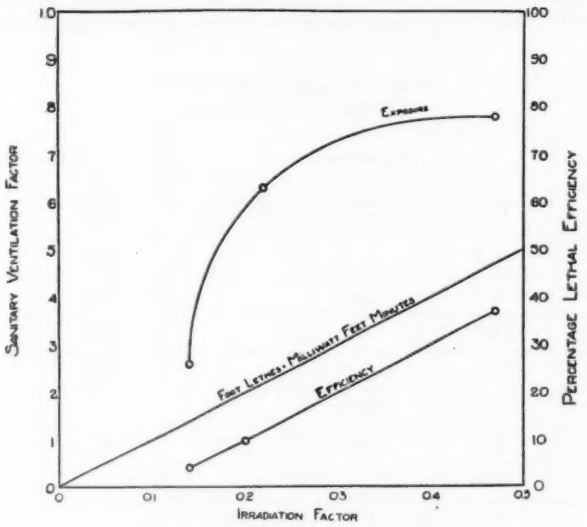

FIG. 1. DISINFECTION FACTORS

Relation between exposure and irradiation factors, and relation between irradiation and sanitary ventilation factors. Exposure factor approaches one as irradiation and sanitary ventilation factors approach one-half because approximately one-half either of the radiant distance of unreflected radiation, or of the total lethal flux absorbed by reflectors, is unavailable for irradiation.

lethal flux emerging from the same fixtures located in the same position in the same room, giving the same uniform cubic flux density, thus gives entirely different average flux density when intensity distance differs because changing angle changes the ray length intercepted by the enclosing surfaces. Ray length is thus as important in irradiation design as ray strength.

Lethes per minute are a sixtieth of equivalent air changes per hour. Radiant disinfection obviously is not determined solely by the quantity of non-uniformly distributed radiation. Thus, lethal change due to irradiation of recirculated air depends primarily upon rate of recirculation, which cannot be exceeded, regardless of the quantity of radiation applied in a duct. Circulation between more and less strongly irradiated zones in a room likewise determines lethes

per minute. The lethal effect of a design must be determined bacteriologically in derivation of disinfection factors. Air motion and equivalent air change offer problems in ventilating engineering.

The irradiation factor is derived by dividing average flux density by uniform cubic flux density. Since the irradiation factor varies with radiation footage, it measures effectiveness of design. Conversion of radiation into irradiation is not the responsibility of the doctor or the lamp manufacturer, but of an irradiation engineer who can unite ventilating, illuminating, and sanitary engineering into one hygienic objective.

The exposure factor is derived by dividing lethes per minute by the average flux density. This also is found not to be directly proportional to the irradia-

TABLE 2—LETHES PER HOUR[a]
(*Between Various Points*)

POSITION	SWARTHMORE			GERMANTOWN	
	Bench Level	Upper Level	Reflector Horizontal	Bench Level	Upper Level
A–1–1	61.0	...	...	30.5	...
A–1–2	91.7	...	...	35.9	...
A–1–3	120.0	...	...	47.6	...
A–1–4	...	...	...	37.5	...
B–2	32.8	1054	...	...	...
B–1	52.9	293	635	36.7	80.0
B–3	75.0	...	...	...	...
B–4	42.9	...	352	...	...

[a] Wells, W. F. and Green, R. Bacteriologic Study of Radiant Disinfection of Air in the Control of Epidemic Spread of Contagion. (Paper read before American Public Health Association, Oct. 14, 1941.)

tion factor (Fig. 1), though dependent upon it in two ways. First, the more nearly irradiation becomes uniform through an enclosed space, the greater will become the exposure factor, regardless of air circulation. Uniformity of exposure of organisms circulating from larger, more irradiated, through smaller, less irradiated, portions of a room, is a second, more important advantage. The former is a problem in light distribution, but uniformity of exposure due to circulation is largely a problem in ventilating engineering.

The sanitary ventilation factor becomes the product of the irradiation and exposure factors. It approaches a half rather than one because approximately one-half, either of the radiant distance of unreflected radiation, or of the total lethal flux absorbed by reflectors is unavailable. Exposure of the test organism to a uniform intensity of a milliwatt per square foot would yield a lethe per minute if irradiation and sanitary ventilation factors are equal. Actual determinations are not yet accurate enough to establish the

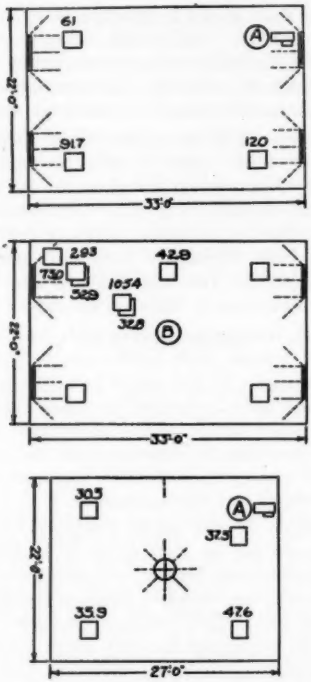

FIG. 2. VENTILATION TESTS OF SCHOOLROOMS AT GERMANTOWN (CENTRAL LIGHT FIXTURE) AND SWARTHMORE (TWO WALL FIXTURES AT EACH END)

A and *B* represent positions of infector, or atomizer, squares represent positions of infectees, or air sampling centrifuges. Double squares represent pairs of machines, one sampling from upper level and one from bench level. Figures indicate lethes per hour between infector and infectee; air circulation counterclockwise.

biophysical constant relating physical and bactericidal units, but the factor still offers a practical means of defining radiant disinfection. Sanitary specifications can be translated by this factor into irradiation.

Foot lethes per occupant per minute express equivalent sanitary ventilation. Organisms circulating between two points would be removed at the same rate as from a room by replacement with an equivalent number of cubic feet of pure air per occupant per minute. The order of magnitude of these values,

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however, is much higher than would be practicable by actual air change during cold weather. Values hitherto unattainable become practicable where rigid requirements demand and disciplinary measures permit protection of the occupants against over-exposure to radiation. Interpretation of infection and disinfection in terms of air conditioning is a sanitary problem.

Lethes per hour between different points in a room, shown on Table 2, take on added significance in the light of these considerations. More uniform distribution of light at Germantown compensates in large measure for the lower uniform cubic flux density (Fig. 2). Correspondingly greater uniformity of the overturn values at different stations reflects the uniformity of radiation distribution. Less variability between bacterial concentrations at upper and lower levels in the two installations also provides an empirical index of the ratio of circulation to disinfection rates.

Correlation of sanitary ventilation factors with irradiation factors, and foot lethes per occupant per minute with epidemiologic indices, will be required for codification. The purpose of this paper is to point out the nature of the data and the kind of instrument required for this purpose, and by presentation of actual data obtained under actual conditions of sanitary ventilation where hygienic values are clearly demonstrated, to emphasize the need of extending such methods to larger problems. Hygienic interpretation of sanitary ventilation is a problem for the sanitarian.

The sanitary significance of the actual values in the control of epidemic spread of contagion should not be misapplied to problems based on different premises, such as prevention of hospital cross-infection. Reduction of the hazard of breathing nuclei from droplets expelled by presumably well persons congregating with other presumably healthy members in an enclosed atmosphere is not the same as protecting the sick from breathing infected dust in a hospital ward, or preventing air-borne infection of a surgical wound. An epidemic wave of contagion, due to prolonged but attenuated exposure in a form of infection highly vulnerable to irradiation may be checked by measures which could not guarantee complete prevention of sporadic cases. These are public health problems.

Air disinfection in surgical antisepsis has been fully demonstrated, but different methods of application and different techniques of evaluation apply to prevention of bacteria-laden dust or droplet nuclei settling into a wound. Design of ultraviolet barriers in the prevention of dust-borne hospital cross-infection in wards or nurseries require different specifications. The flux density traversed by a dust particle passing from one cubicle to another must be far greater than would be sufficient for the sterilization of droplet nuclei. These are problems in hospital administration.

Modifications in technique, however, should not obscure the fact that ventilating engineers must seriously consider hygienic obligations imposed by any accepted theory that enclosed atmospheres constitute vehicles for the spread of infection. They but further multiply reasons for study of air-purification methods which promise to eliminate hygienic hazards from air supplies. With a country looking forward to a long struggle between massed populations, the problem of controlling epidemic spread of respiratory infections will become acute. Army concentrations within controllable atmospheres present special problems.

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ACKNOWLEDGMENT

These collaborative studies between the School Boards of the Germantown Friends School and the Swarthmore Public Schools, the Hanovia Chemical and Manufacturing Co., the General Electric Co., and the Laboratories for the Study of Air-borne Infection, were made possible through the efforts of Dr. A. N. Richards, vice-president of the University of Pennsylvania, Dr. Joseph Stokes, Jr., professor of Pediatrics, University of Pennsylvania, Stanley Yarnall, principal of the Germantown Friends School, and Frank R. Morey, principal of the Swarthmore Public Schools. Bacteriologic data obtained by Richard Green, research associate, with the assistance of Miss Dorothy Wells, were also applied, and photometric measures were made by Arthur Hammerlund.

DISCUSSION

DR. HOLLAENDER: The author has done a great deal of pioneer work on the control of air-borne contagion. However, I feel that it would be helpful if a definition of bacteriocidal radiation would be stated in cooperation with the Council on Physical Therapy of the *American Medical Association,* who have just published an acceptance of the use of ultraviolet lamps for disinfecting purposes. It is extremely difficult to define the lethe, or accept it as Professor Wells has defined it, especially since it is not simple to have standard suspension of an organism. This also applies to the other definitions which Professor Wells has given.

PROFESSOR WELLS: I think Dr. Hollaender's insistence on complementing the report of the Council on Physical Therapy is quite proper. The Council has recently defined the strength of burners which are acceptable to the *AMA,* but disclaims the responsibility, if I understand their report, for defining dosage which determines the breathing quality of confined atmospheres. My paper attempts a system of units by which dosage can be defined—a first attempt, but one necessary before reporting upon clinical results of radiant disinfection.

It is much more simple to determine the dose of a drug by the amount in a pill or a spoonful, and it is not difficult to compute or to test the concentration of a chemical disinfectant in the atmosphere. It is more difficult to specify the density of a barrage, though this also is necessary if you are to prevent penetration. However, the physics of ultraviolet emission can be described and the geometry of irradiation between a source and a barrier which absorbs the lethal photons can also be accurately specified. Lethal uniformity of radiation in space cannot be defined with physical precision, nor can the individual paths of drifting bacteria be predicted. A statistical average can however be specified just as pressure specifies the statistical average of molecular impingements. The paper attempts to formulate the results of these statistical integrations.

It is also quite true that standard suspension of standard organisms is not a physically or chemically perfect procedure. Standardization of disinfectants has long been a subject of argument among bacteriologists, due to the nature of biological processes and phenomena. However, it was necessary to standardize disinfectants for intelligent application, and it will be equally necessary to standardize radiant disinfection. Hair-splitting details have not prevented progress in air-conditioning, nor should they impede progress in sanitary ventilation.

The results of radiant disinfection in the schools are evident to every mother of the school children—to the school teachers, the school boards, and others in the communities who have witnessed the experiments. It is not necessary that they go into statistical details on immunity or susceptibility, or await the pronouncements of a super-board of experts. It was obvious that during the largest epidemic of measles in

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Philadelphia the unirradiated upper grades suffered far more heavily than the irradiated lower grades. In previous years in these schools measles has attacked the primary school age (characteristic of measles since the dawn of medicine)—five or six in these ages for one in High School ages. For the first time the age incidence of measles has been reversed.

This experience convinces us that ventilating engineers stand at the threshold of a new phase of their art. Just as it was necessary for water supply engineers to absorb sanitary specifications into their design, so has it become necessary for ventilating engineers to accept hygienic specifications. Engineers must play a part equal in importance to that assumed by the medical profession in the pronouncement of the Council on Physical Therapy.

No. 1211

PRESSURE LOSS CAUSED BY ELBOWS IN 8-INCH ROUND VENTILATING DUCT

By M. C. STUART,* C. F. WARNER,** W. C. ROBERTS,*** BETHLEHEM, PA.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with Lehigh University.

INTRODUCTION

A LARGE per cent of the total pressure loss experienced in ventilating systems is caused by elbows. If this loss can be reduced, more efficient systems may be designed. In 1939 the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS entered into a cooperative agreement with Lehigh University for the purpose of studying the pressure loss due to elbows. The program will include the study of all types of elbows for 8-in. round and 7 x 7 in. square and other size ducts. This paper presents the study of round elbows in 8-in. duct. The 7 x 7 in. square duct is now under investigation.

PRESSURE DROP IN STRAIGHT DUCT

Apparatus for the study of pressure drop in straight duct was set up as shown in Fig. 1A. Duct having 40 joints per 100 ft was used with all joints taped to prevent leakage. Air supplied by a fan having a capacity of 3000 cfm at the pressures desired, discharged into a box which broke up the circulation caused by the fan so that normal turbulence was maintained in the test section. An egg crate straightener was used to further insure normal turbulent flow. The duct discharged into a 30-in. diameter plenum chamber fitted with a calibrated discharge nozzle. Static tube pressure taps were placed every 10 ft along the test section. The reasons for the use of the static tube may be found in Appendix I. These taps were connected differentially through commercial single inclined tube manometers. The last pressure tap before the plenum chamber was connected to a manometer having

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the other leg open to the atmosphere. An over-all pressure differential was taken to be used as a check on the small differentials.

The velocity was obtained by dividing the quantity of air flowing by the area of the duct. All calculations were based on an assumed nominal diameter of 8-in. for the duct. This is not the exact diameter since so-called 8-in. duct has a diameter which varies from 7⅞ in. to 8⅛ in. For this reason the velocity used was an average nominal velocity. (See Appendix III for derivation of velocity equation.) The quantity of flow was varied by throttling the inlet of the fan.

The results of the investigation on pressure loss for 100 ft of straight duct are shown in Fig. 3 by the curve labeled *Straight Duct*. The straight duct

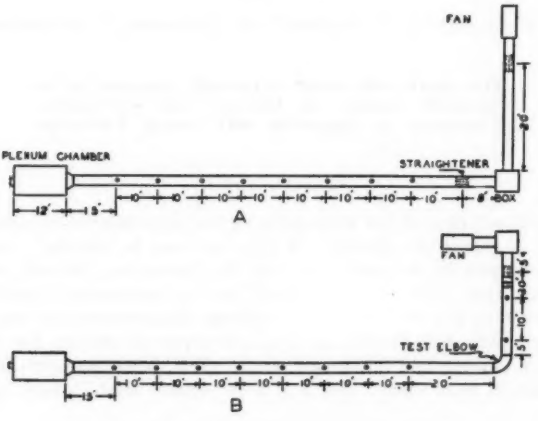

FIG. 1. ARRANGEMENT OF APPARATUS

a. For study of friction drop in straight duct (40 joints per 100 ft)
b. Test set-up for determination of pressure drop around an elbow (nozzles used on plenum chamber have throat diameters of 4 and 8 in.).

pressure loss is being investigated by the Society's Pittsburgh Laboratory, but, as will be shown later, these values are needed in the present investigation of elbow pressure loss. The results [1] of this investigation check with those of the Laboratory as presented in the paper, previously published.

PRESSURE DROP IN A SINGLE ELBOW

Apparatus for the study of pressure drop in a single elbow was that used in the straight duct test but modified as shown in Fig. 1B. An elbow is defined as the section of duct joining two straight lengths of duct. Three

[1] Analysis of the Factors Affecting Duct Friction, by J. B. Schmieler, F. C. Houghten and H. T. Olson. (ASHVE TRANSACTIONS, Vol. 46, 1940.)

pressure taps preceding the elbow and three following were used. At various times during the test additional taps were used as a check on the work.

The results of the investigation of pressure loss in a single elbow are shown

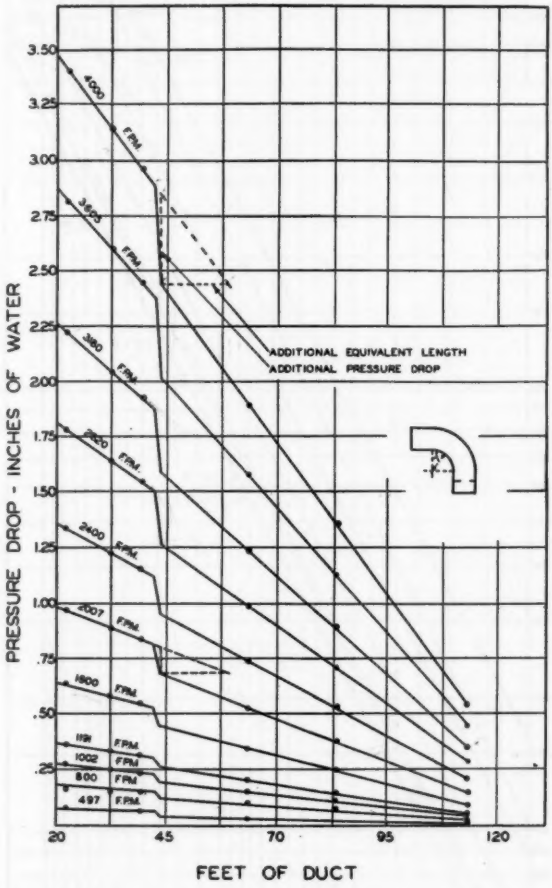

FIG. 2. RELATION BETWEEN PRESSURE DROP AND VELOCITY FOR A 7-IN. INSIDE RADIUS ELBOW (NOTE THAT ADDITIONAL EQUIVALENT LENGTH DOES NOT CHANGE WITH VELOCITY)

in Fig. 2, a plot of pressure drop *vs.* length along the duct for different velocities. Pressure loss in an elbow is caused by three flow phenomena: (1) the friction of the air particles against the duct wall; (2) the loss due to turbulent flow; and (3) the loss due to a change in direction of flow.

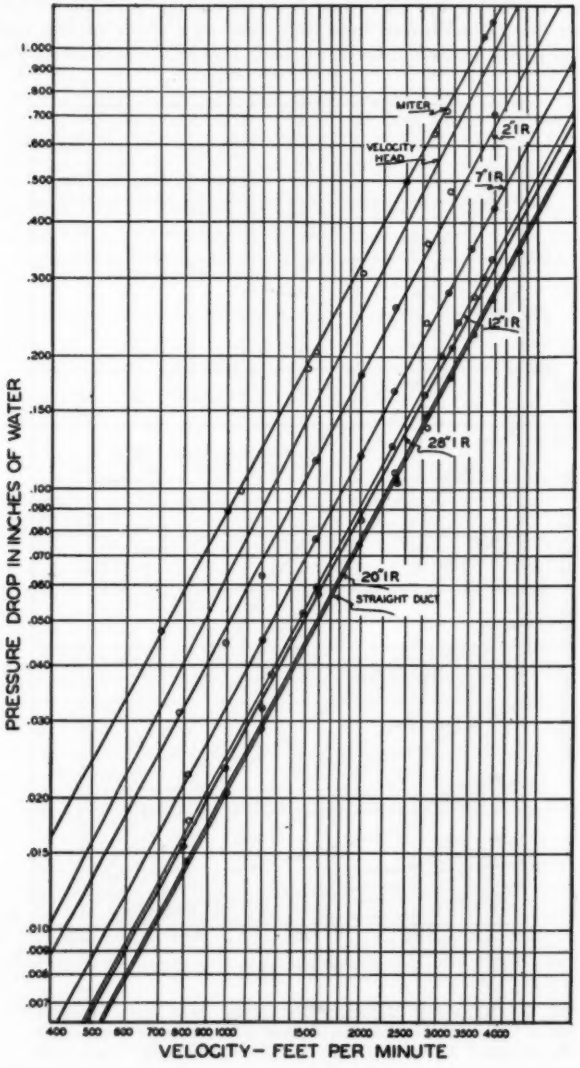

FIG. 3. RELATION BETWEEN TOTAL PRESSURE DROP AND VELOCITY FOR ALL ELBOWS TESTED. THE CURVE, STRAIGHT DUCT, IS FOR 10 FT OF DUCT WITH 40 SLIP JOINTS PER 100 FT

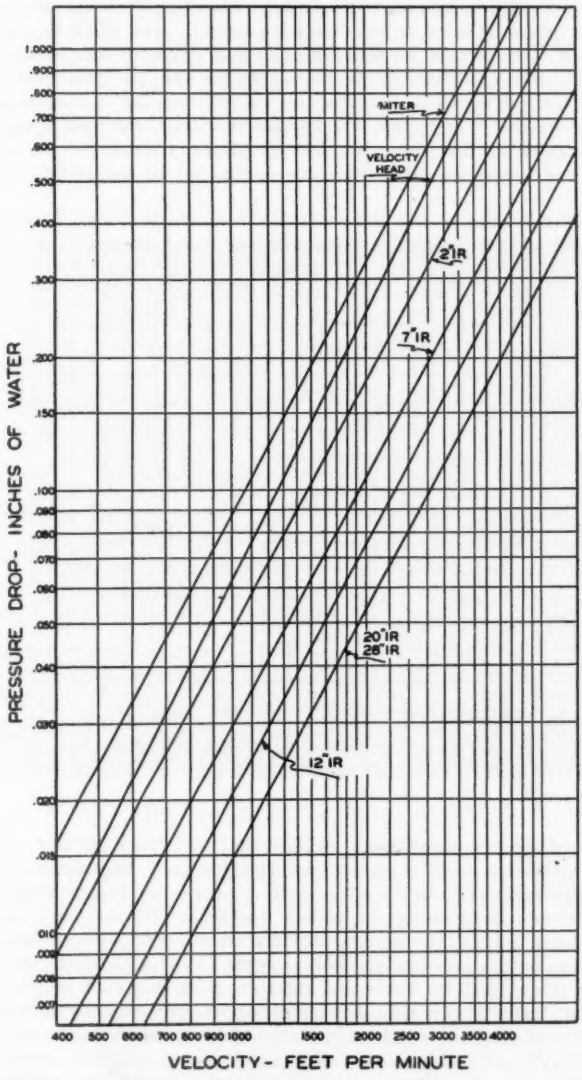

FIG. 4. RELATION BETWEEN ADDITIONAL PRESSURE DROP AND VELOCITY FOR ALL ELBOWS TESTED

Additional pressure loss is defined as the excess of measured pressure drop for a section of duct including an elbow over the calculated pressure drop for a section of straight duct having an equal centerline length.

The first two items constitute the loss experienced in straight duct and also occur in an elbow because it has a definite length. The third loss is unique to the elbow, and may be termed the *additional pressure loss caused by the elbow. This additional loss is determined by finding the excess of pressure drop for a section of duct including an elbow over the calculated pressure drop for a section of straight duct having an equal centerline length.* The total loss caused by the elbow is made up of the friction loss based on its centerline length and the additional loss determined by the method described herein.

Examination of Fig. 2 will show that the pressure gradients preceding and following the elbow are parallel straight lines but some distance apart. If

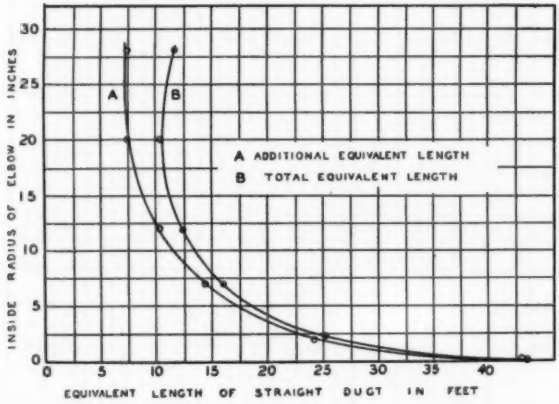

FIG. 5. RELATION BETWEEN INSIDE RADIUS OF ELBOWS AND EQUIVALENT LENGTH OF STRAIGHT DUCT AT ANY VELOCITY

the length of duct involved were straight and the elbow omitted, points on a pressure gradient would be on the same straight line. Because of the abnormally high pressure drop in the elbow with respect to its length, a line indicating the pressure drop along the center line length of the duct and elbow shows a discontinuity. The vertical distance on the pressure axis between the two segments of the pressure gradient is equal to the additional pressure loss due to the elbow, and the horizontal distance along the duct length axis between the segments is precisely that length of straight duct which might have been added to the elbow length to cause an equal pressure drop. Since there is within the elbow itself a friction drop admittedly proportional to the centerline length of the elbow, we may add this friction loss to the additional loss and term the result the total loss due to the elbow. Similarly, the length of the elbow added to the additional equivalent length may be termed the total equivalent length.

Fig. 2 also shows that for a given elbow the additional equivalent length

does not change with velocity, consequently the total equivalent length will not change. For a given elbow a unique length of straight duct may be substituted causing a pressure loss exactly equal to the loss due to the elbow. It immediately follows that the loss in an elbow is the same power function of velocity as the loss in straight ducts. Equivalent length of duct is of more significance than pressure loss because as was shown it is independent of velocity. Additional equivalent length, the excess of pressure drop for a section of duct including an elbow over the calculated pressure drop for an

FIG. 6. ALL OF THE ELBOWS STUDIED

equivalent centerline length of duct divided by the pressure loss per foot of duct, is for some purposes of more importance than total equivalent length.

EFFECT OF VARYING RADII ON PRESSURE LOSS IN ELBOWS

Elbows with radii from 0 to 28 in., shown in Fig. 6, were investigated in the arrangement shown in Fig. 1B. The procedure for measuring the pressure loss and velocity already described was used for all elbows. The curves of total loss *vs.* velocity are given in Fig. 3. A plot of additional loss *vs.* velocity is given in Fig. 4. Examination of the curves will show that the total pressure loss decreases with increasing radii. When an optimum value has been reached, further increases in radii will increase the total pressure loss. The additional pressure loss decreases with increasing radii. It may

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be concluded that as the radius increases the friction loss becomes the dominant value.

The numerical relationship existing between elbows of different radii and equivalent lengths of straight duct is shown in Fig. 5. The additional equivalent length of elbows, Curve *A*, of Fig. 5, decreases with increasing radii and very nearly approaches a constant value at large radii. Also in Fig. 5 Curve *B* of the total equivalent length of straight duct reaches a minimum value and then increases with increasing radii.

From the results the important conclusion reached is that in elbows of large radii, the pressure loss caused by changing the direction of flow is of minor

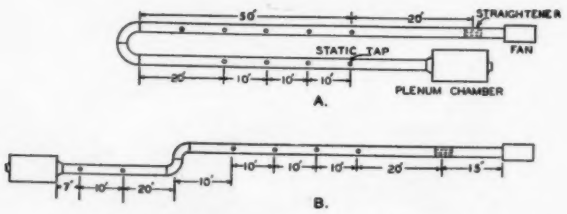

FIG. 7. TEST ARRANGEMENT OF APPARATUS

a. For the study of pressure loss caused by a 180 deg change in the direction of flow
b. Arrangement of apparatus used in the study of offsets (nozzles used on plenum chamber have throat diameters of 4 and 8 in.)

importance compared to the pressure loss caused by friction and turbulence over its centerline length.

ELBOWS IN SERIES

Two elbows separated by a length of straight duct 40 diameters long were investigated to determine the effect of one elbow on another farther down the duct. No effect of the first elbow on the second elbow was observed; therefore, it may be concluded that if the distance separating the elbows is greater than 40 diameters one elbow will not affect another.

An investigation was made, using the set-up shown in Fig. 7A to determine the loss of 180 deg bends made up of two 90 deg elbows. The bends, shown in Fig. 8 were tested in the same apparatus used previously. As a result of this study it may be concluded that the total pressure loss in 180 deg bends is never as much as that of two elbows and decreases as the radius increases. In fact for large radii the loss in 180 deg bends is scarcely more than that in the 90 deg elbow.

The same effect was noted when the elbows, installed as offsets, shown in Fig. 9, were tested as arranged in Fig. 7B. From these studies it may be concluded that after the direction of flow has once been changed, further change in direction of flow does not increase the loss materially. The one exception to this is the miter elbow. When two of these are used together, the total loss is twice that of one.

FIG. 8. U BENDS MADE UP OF TWO 90 DEG ELBOWS OF THE SAME INSIDE RADIUS

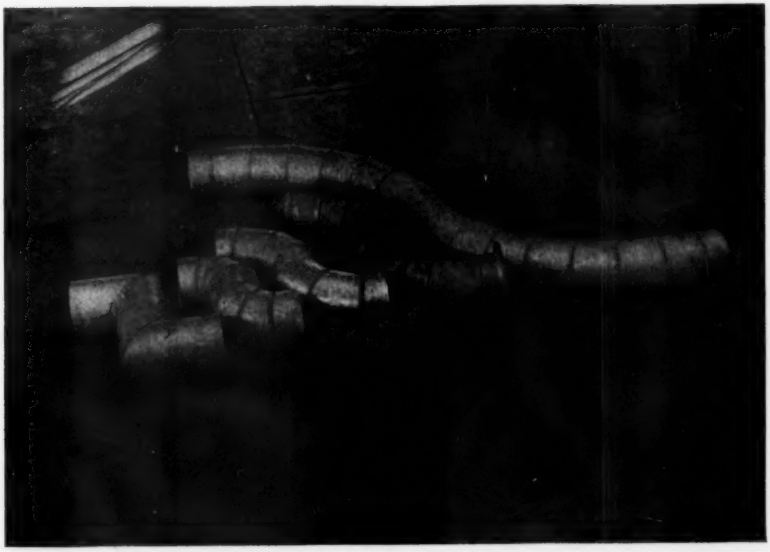

FIG. 9. THE SAME ELBOWS AS SHOWN IN FIG. 8 BUT ASSEMBLED TO FORM OFFSETS

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VANED ELBOWS

In the 8-in. duct two elbows, the miter and the 2 in. inside radius elbows, were fitted with shop made splitters fastened to a threaded rod extending from the inside to the outside of the duct 45 deg around the bend. The rods were movable so that the splitters could be located any place across the elbow. Splitters [2] of different heights and lengths were tried in various positions across the duct until the most effective splitter and most efficient position for it were found. Tests were then made to determine the pressure loss, the same method being used as was employed in determining the pressure loss of a single elbow.

It was found that one splitter, 4½ in. long, 7 in. high, bent on a 3½ in. radius, and located 1½ in. from the inside wall of the 2 in. inside radius elbow, gave the lowest pressure drop and a reduction of 20 per cent in the total equivalent length. A reduction of 40 per cent in total equivalent length of the miter elbow was obtained by one splitter 6½ in. long, 7¼ in. high, bent on a 4 in. radius, and located 2¼ in. from the inside wall. No appreciable decrease in loss was noted when the splitters were located near the outside wall of the elbows. No material reduction in pressure loss was obtained by the use of two splitters instead of one in the miter elbow.

CONCLUSIONS

This paper presents a method of determining experimentally the pressure loss caused by elbows in ducts of ventilating systems.

The data presented include results of the investigation of pressure loss caused by various types of elbows in 8-in. round duct.

Significant conclusions arrived at from the data presented include the following:

1. The pressure loss caused by elbows is proportional to the 1.84 power of the velocity.
2. The pressure loss of an elbow at any velocity may be expressed by a unique equivalent length of straight duct.
3. Splitters reduce the pressure loss in elbows of small radii. One splitter in a 2-in. inside radius elbow reduces the loss 20 per cent; one in the miter elbow reduces the loss 40 per cent.
4. Equivalent length of pipe for elbows having varying radii as follows:

INSIDE RADIUS	ADDITIONAL EQUIV. LENGTH IN FEET	CENTERLINE LENGTH IN FEET	TOTAL EQUIVALENT LENGTH IN FEET
0 Miter	42.9	0.6	43.5
2 in.	24.1	0.8	24.9
7 in.	14.5	1.5	16.0
12 in.	10.3	2.1	12.4
20 in.	7.5	3.1	10.6
28 in.	7.2	4.2	11.4
0 Miter (vaned)	25.7	0.6	26.3
2 in. (vaned)	19.3	0.8	20.1

[2] See Appendix IV for description of splitters tested.

APPENDIX I

DISCUSSION OF STATIC PRESSURE DIFFERENTIAL MEASUREMENTS

In order to select the most suitable means of measuring static pressure differential, four methods were tested, *viz:* piezometer ring, single impact tube, averaging impact tube, and single static tube.

First, piezometer rings were spaced every 10 ft and connected differentially. When the results were plotted it was found that the pressure gradient did not satisfactorily fulfill the condition of uniform slope. This deficiency may have been caused by local conditions at the piezometer ring due to the proximity of the ring to the duct joint.

Tests were next made to study the possibility of the use of impact tube, averaging impact tube, and static tube. For this study a straight section of duct, 80 ft long, was set up as shown in Fig. 10 with three surveying stations 15 ft apart. Each station was equipped with holes, to permit insertion of survey tubes, and a piezometer ring so located that local conditions could not affect its use. Three different survey tubes, each 3/16 in. inside diameter, were

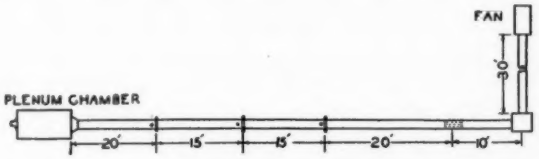

FIG. 10. TEST APPARATUS FOR THE COMPARISON OF PRESSURE MEASURING DEVICES (THE NOZZLE OF PLENUM CHAMBER HAS AN 8-IN. THROAT)

used. No. 1, the impact tube, had an upstream length of 4 in. terminating in a well-rounded entrance. No. 2, the averaging impact tube, was made by drilling nine 0.04-in. holes ¼ in. apart in a straight line over a 2-in. length in the middle of a tube 20 in. long and closed at one end. When used the tube extended across the duct at its diameter with the holes pointed upstream parallel to the direction of flow. No. 3, the static tube, ASHVE type, had an upstream length of 4 in. with four 0.04-in. holes drilled at right angles to each other 1½ in. from the rounded end. In making surveys, the tubes were connected differentially through a manometer to the piezometer ring.

The surveys were made at all three stations at velocities of 3600, 3000, 1,000, and 500 fpm. In all, about 300 readings were taken. The results are shown in Figs. 11, 12, and 13. Examination of Fig. 11 will show that the impact tube does not give satisfactory results when used as a fixed instrument. Fig. 12 shows that the averaging impact tube gives more uniform pressure patterns but is still unsatisfactory except at very low velocities. Fig. 13 showing static pressure versus piezometer ring static pressure indicates that neither velocity nor location along the duct affects the result.

The preceding discussion indicates that the static tube gives the most reliable results since it is not affected by location. For this reason and because of its simplicity in installation and use, static tubes fixed at the geometric center

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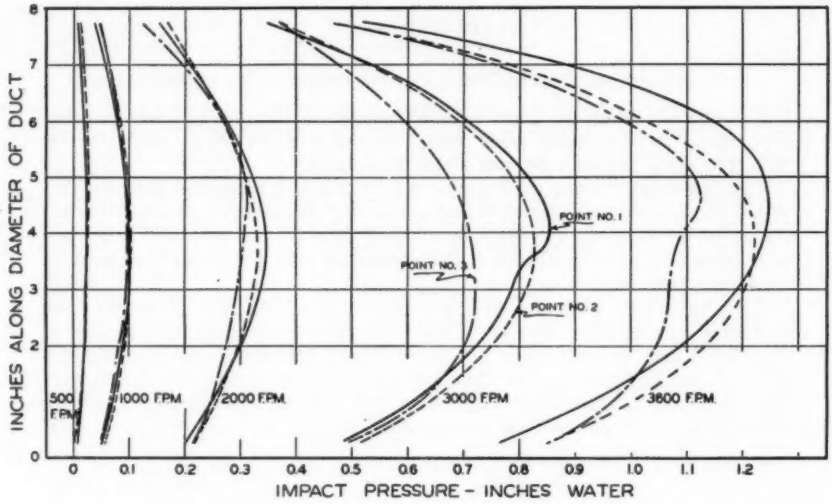

FIG. 11. RELATION BETWEEN POSITION ALONG DIAMETER OF DUCT AND IMPACT PRESSURE, AS MEASURED BY A ⅛ IN. BORE IMPACT TUBE, FOR DIFFERENT VELOCITIES

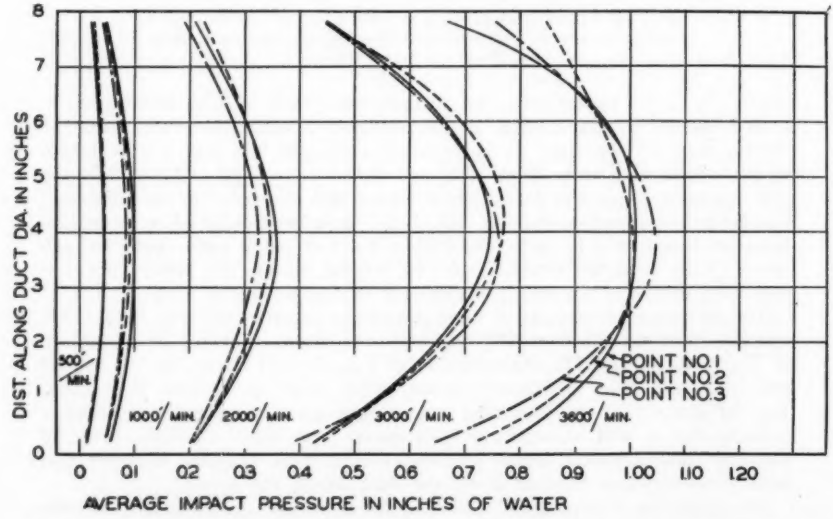

FIG. 12. RELATION BETWEEN POSITION ALONG THE DIAMETER OF DUCT AND IMPACT PRESSURE, AS MEASURED BY AN AVERAGING IMPACT TUBE AT DIFFERENT VELOCITIES

of the duct were used to make all pressure measurements made in this investigation.

APPENDIX II

THE LEHIGH MICRO-MANOMETER

To insure accurate results, a calibrating manometer, Fig. 14, was developed and constructed. This manometer is patterned after one made by Professor

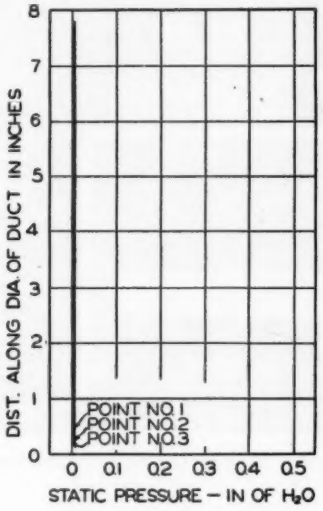

FIG. 13. RELATION BETWEEN POSITION ALONG THE DIAMETER OF DUCT AND PRESSURE DIFFERENTIAL BETWEEN A STATIC TUBE INSIDE THE DUCT AND THE PIEZOMETER RING STATIC PRESSURE, AT VELOCITIES FROM 3600 TO 500 FPM

Merriam of Worcester Polytechnic Institute. The manometer consists of two 3½-in. diameter glass cylinders ground to the same inside diameter and connected by a small brass tube. One cylinder is closed and is connected to the gage being calibrated. The other cylinder, open to the atmosphere, is fitted with an indicating pointer actuated by a micrometer screw. The micrometer screw was made for the manometer and is accurate to 0.00015 in. in 5 in. The micrometer head is so divided that readings are made to the nearest thousandth and estimated to the nearest ten-thousandth of an inch. Elimination of specific gravity determinations is obtained by using distilled water in the manometer.

The instrument is used by connecting one leg of the manometer to be calibrated to the closed cylinder of the micro-manometer. The other end of the manometer is open to the atmosphere as is the remaining cylinder of the micro-manometer, thus insuring a uniform reference pressure. When equilibrium

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has been established, the pointer is brought from above into contact with the water surface in the micro-manometer and a reading is taken. Pressure is then applied to the tube connecting the two manometers and again readings are taken. The difference in readings of the micro-manometer multiplied by two is the correct pressure in inches of water.

Indication of contact between the pointer and the water surface is given by a vacuum tube ammeter electric circuit. At the instant of contact the ammeter

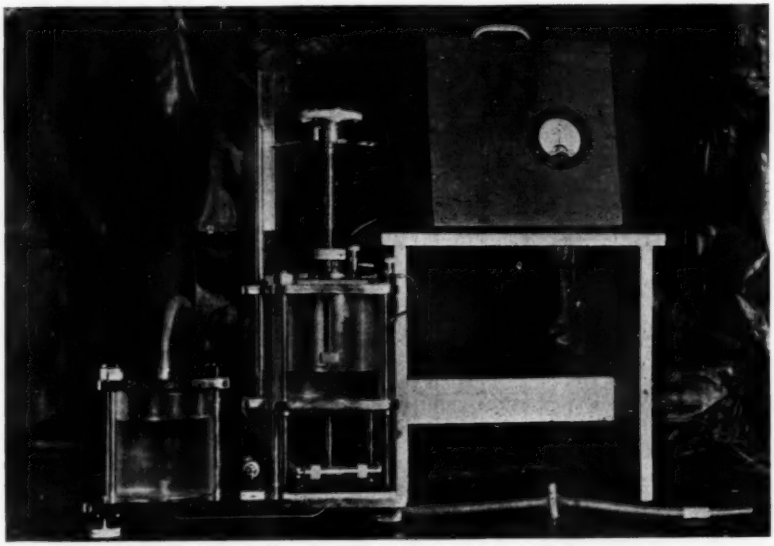

FIG. 14. THE LEHIGH MICRO-MANOMETER WITH ELECTRICAL CONTACT INDICATION

REFERENCES

shows a large deflection. The instant of contact is indicated in this manner more nearly correctly than can be achieved by observing the pointer as it approaches the water surface.

APPENDIX III

VELOCITY CALCULATIONS FOR NOZZLE ON TEST SET-UP

Notations:

H = head, feet of air
h = head, inches of water
g = 32.2
Q = quantity of air flowing
d = density of air, lb/ft³
V_n = velocity in nozzle, fpm
V_d = velocity in duct, fpm
A_n = area of nozzle, ft²
A_d = area of duct, ft²
D_n = diameter of nozzle
D_d = diameter of duct
C = nozzle coefficient = 0.99

General Nozzle Equation:

$$V = \sqrt{2gH}$$

$$V = \sqrt{2 \times 32.2 \times \frac{62.4}{d} \times \frac{h}{12}}$$

$$V = 18.3\sqrt{h/d}$$

$$V_n = 60\ CV = 1096.5\sqrt{h/d}$$

$$V_n = 1084.5\ \sqrt{h/d}$$

$$V_d = Q/A_d$$

$$V_d = V_n\ (A_n/A_d)$$

$$A_n/A_d = (D_n/D_d)^2$$

$$V_d = 1084.5\sqrt{h/d}\ (D_n/D_d)^2$$

APPENDIX IV

Listed as follows are the dimensions of the shopmade splitters which were tested in the miter elbow and the two-inch elbow.

FOR THE MITER ELBOW:

VERTICAL HEIGHT	LENGTH	RADIUS
7½ in.	5½ in.	2½ in.
7½ in.	4⅜ in.	2½ in.
5¼ in.	4¼ in.	2½ in.
6 in.	7 in.	4 in.
6⅜ in.	5¾ in.	3¾ in.
7¼ in.	6½ in.	4 in.
7⅛ in.	7⅞ in.	4 in.
4⅜ in.	2¾ in.	1⅝ in.

FOR THE STANDARD (TWO-INCH RADIUS) ELBOW:

4⅜ in.	2¾ in.	1⅝ in.
7 in.	3 in.	1¾ in.
7 in.	4½ in.	3½ in.
6¾ in.	4¼ in.	2½ in.
7 in.	7 in.	4 in.
7 in.	5¾ in.	3¼ in.

REFERENCES

Loss of Pressure Due to Elbows in the Transmission of Air Through Pipes or Ducts, by F. L. Busey. (ASHVE TRANSACTIONS, Vol. 19, 1913.)

Frictional Resistance to the Flow of Air in Straight Ducts, by F. C. Houghten and J. B. Schmieler. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

Pressure Losses in Rectangular Elbows, by Madison and J. R. Parker. (*ASME Transactions*, Vol. 58, No. 3, April 1936.)

New Data for the Design of Elbows in Duct Systems, by L. Wirt. (*General Electric Review*, Vol. 30, 1927.)

Winter Air Conditioning. (*National Warm Air Heating and Air Conditioning Association*, 1939.)

ASHVE HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1940, Chapter 30.

Fundamental Constants for Engineering Computations with Air, by Sanford A. Moss. (Reprint *General Electric Review*, August 1931.)

Engineering Computations for Air and Gases, by Sanford A. Moss and Chester W. Smith. (*ASME Transactions*, APM-52-8.)

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DISCUSSION

QUESTION: What is the minimum radius of elbow which is safe without having the pockets?

MR. WARNER: We have found that a 2-in. inside radius is about the smallest we can get for an 8-in. circular duct.

A. P. KRATZ, Urbana, Ill.: I think most of our experimental work is indicated on the basis of inside radius which has much more bearing on the pressure loss through an elbow than the outside radius. Some of your slides seem to indicate exactly the reverse, which I would appreciate having you reconcile.

MR. WARNER: The pictures were taken with square 7-in. ducts, and round on the outside. Previous investigations show about a 20 per cent reduction in pressure loss by having the outside square but we could not obtain this result in our test. We used one which was square on the outside, with a 2-in. inside radius that had approximately the same loss as that with the radius on the outside.

QUESTION: In the last picture shown has the effect of splitters been investigated?

MR. WARNER: This phase of the investigation has not been completed, but we are now installing curved vanes in the model. The pressure reduction in square ducts with vanes seems to be quite large as compared to round ducts. We have some data which indicated that we can reduce the loss as much as 65 per cent with the proper introduction of turning vanes in a square duct.

No. 1212

OBSERVED PERFORMANCE OF SOME EXPERIMENTAL CHIMNEYS

By R. S. DILL,* P. R. ACHENBACH ** AND J. T. DUCK,† WASHINGTON, D. C.

INTRODUCTION

PUBLISHED information [1] concerning draft and temperature conditions and friction losses has been limited for the most part to large stacks or chimneys such as those used in industrial power plants. In these larger systems reliable information about stack performance is obviously desirable because of the considerable financial savings that may be involved, and large scale investigations are more practical for industrial organizations than for individuals who operate small plants or heating systems. Even for larger chimneys the experimental data available are limited.

Several types of chimneys of domestic sizes have been constructed and their performance observed at the National Bureau of Standards for the reason that experimental data on small chimneys were not available. The work was done in connection with the low cost housing program since the subject is important in that field.

It is the object of this paper to present the more important data gathered on brick chimneys. Time has not permitted any exhaustive theoretical analysis of the results of the tests but the results are presented for their inherent interest.

When tests for brick chimneys were being considered, it was planned to construct one or more such chimneys and to supply them with air heated electrically in lieu of actual products of combustion from a fire in order to minimize errors of observation and computation. There was a difficulty, however, in obtaining electrical equipment with which to accomplish this and some consideration indicated that the method finally selected would be satisfactory.

This method consisted essentially in heating air for supply to the chimney with a gas flame in an insulated fire box so that substantially all the heat of

* Chief, Heat Transfer Section, National Bureau of Standards. MEMBER of ASHVE.
** National Bureau of Standards.
† National Bureau of Standards.

[1] Determination of Chimney Sizes, by Alfred Cotton. (*Mechanical Engineering*, September, 1923.)
Proportioning Chimneys on a Gas Basis, by Menzin. (*ASME Transactions*, 1915.)

Presented at the 48th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January, 1942.

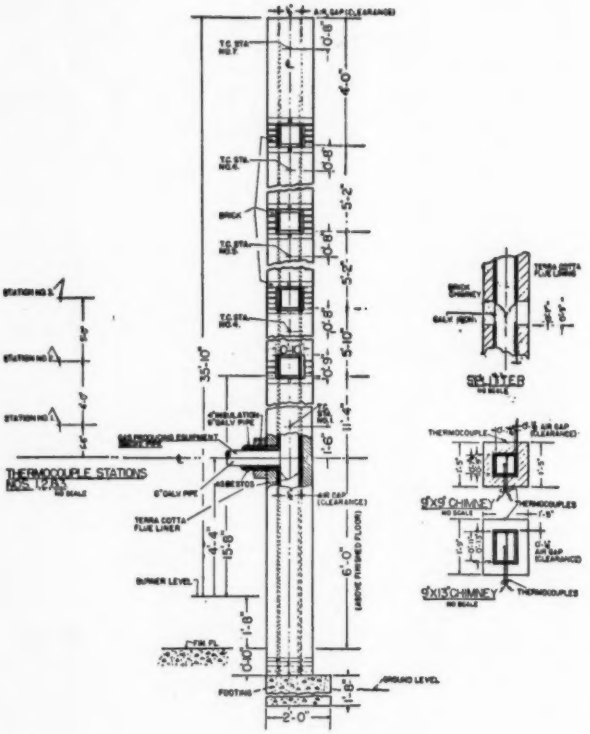

FIG. 1 (*left*). MASONRY CHIMNEY WITH TERRA COTTA FLUE LINING

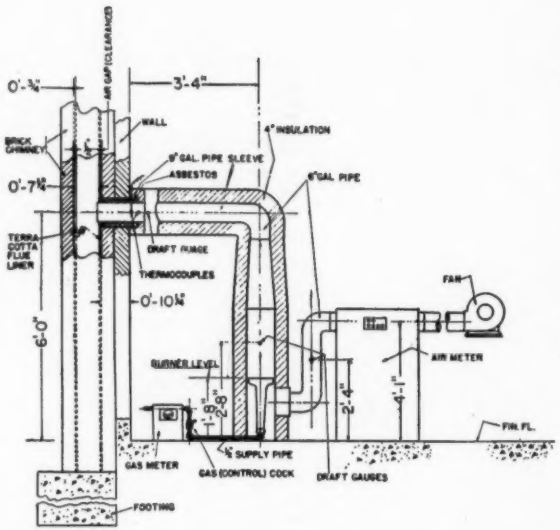

FIG. 2 (*above*). FLUE GAS PRODUCING EQUIPMENT FOR CHIMNEY TESTS

combustion was carried to the chimney by the air. The air passing through the system was measured by means of a large gas meter and the gas consumed was measured with a small gas meter so that the rate of mass flow was obtained directly.

Since the resistance to flow in the large meter was considerable, a blower and motor were necessary to overcome it and to cause a sufficient air flow through the meter.

The intent was to determine experimentally the performance of the chimney without reference to any particular kind of heating device and for this reason the flue gas temperature and available draft were measured in the flue pipe close to the chimney.

It was desired to cover the range of operating conditions likely to occur in domestic chimneys, so tests were planned during which chimneys would be supplied gases at 200, 400, 600, 800 and 1000 F. The rates of flue gas production resulting from the combustion of ½, 1 and 1½ gal of fuel oil per hour with 8 per cent and with 10 per cent CO_2 were calculated so that the available draft of a typical chimney could be observed when it was receiving gases at these rates and at the stated temperatures. The work was based on oil consumption for convenience because oil burners are commonly rated in gallons per hour. A correction would obviously be necessary if the data were used for coal burning devices since the gas evolution for the same heat release is somewhat different for that fuel.

The tests were purposely made at times when there was little or no wind in order to avoid that complication.

Data tabulated herewith show the rate at which gases were supplied the chimneys during typical tests:

FUEL RATE	GPH	½		1		1½	
CO_2 in Flue Gases.....	Per Cent Vol.	10	8	10	8	10	8
Gas Flow.....	Cfm (Std.)[a]	18.4	22.5	36.8	45	55.2	67.5

[a] Corrected to 70 F and 29.92 in. Hg.

For computation of the static draft the formula used was

$$D = \frac{d\,H\,(T_1 - T_2)}{5.2\,T_1}$$

in which D = static draft, in. water gage (W. G.)
d = mean density of atmospheric air at time of test, pounds per cubic foot
H = effective height of the chimney, feet
T_1 = mean temperature of gases in the chimney, degree F absolute
T_2 = temperature of air surrounding the chimney, degree F absolute

It was assumed for the purpose of this computation that the flue gas would have the same density as atmospheric air at the same temperature and pressure.

The static drafts were computed for the various conditions employed during these tests and are shown on the tables of results for comparison with the

observed available draft. The results show that the difference between the observed available draft and the computed static draft was not great for the conditions of the tests. An important conclusion is therefore that the draft in a conventional domestic chimney could be computed close enough for practical purposes by means of the previous formula, neglecting friction, if the mean temperature of the gases in the chimney could be known. The results show further, however, that the temperature of the gases decreases greatly when they enter the chimney, even though the chimney had been in operation several hours. A mathematical determination of the temperature conditions in a chimney is undoubtedly complex. It is complicated by the fact that conditions in a domestic brick chimney probably never arrive at a steady state of operation. These things indicate that it is safer and simpler to arrive at the available draft under a given condition by direct experiment rather than to attempt a mathematical solution of the temperature condition in the chimney.

For these reasons the observed relations between chimney dimensions, entering gas temperature and quantity, and available draft are considered to be the more important essentials presented in this paper.

In order to compare the drafts available under the various operating conditions it was desirable to correct all observations to some standard condition and the condition chosen for this standard was an outdoor temperature of 32 F and a pressure of one standard atmosphere. The formula used for making such corrections was $S_2 - S_1 = O_2 - O_1$ *where* S_1 is the computed static draft under the conditions of experiment, S_2 is the computed static draft under standard conditions, O_1 is the observed draft under experimental conditions and O_2 is the corrected draft or draft that should have been observed if test conditions had been standard.

FIG. 3. VIEW OF EXPERIMENTAL CHIMNEYS

This relation develops from the assumption that the friction (f) in the chimney and the gas flow are the same under the experimental and under the standard conditions, because

$$S_1 - O_1 = f \text{ and}$$
$$S_2 - O_2 = f \text{ so that}$$
$$S_1 - O_1 = S_2 - O_2 \text{ or}$$
$$S_2 - S_1 = O_2 - O_1$$

SPECIMENS AND TEST EQUIPMENT

The details of the two brick chimneys used for these tests are shown in Fig. 1. These chimneys were built of common brick, laid in one 4-in. thickness around a tile flue liner. The liner in one chimney was 9 x 9-in. nominal size and that in the other was 9 x 13-in. The measured inside dimensions of the liners were 7¼ x 7¼-in. for the smaller and 6½ x 11-in. for the larger chimney. Each chimney was 38 ft high above the concrete base and each was pierced with vents, 9-in. high and 10-in. wide, opposite each other in two sides of the chimney 15 ft above the burner level and at 5 ft intervals from there to the top. By opening the vents at any level and closing the chimney above them the effective height of the chimney could be established at selected heights less than the whole height of the chimney.

The chimneys were built adjacent to a structure with corrguated metal sides rising to a height of 10 ft but were not built into the wall of the structure. Thus, in effect the chimneys were exposed to the outside air on all four sides for their entire height. A cross section of the chimneys showing the location of the vents and smoke pipe thimble is presented in Fig. 1.

The mortar used was a Portland cement, lime and sand mixture proportioned for one part of cement and lime to about three parts of sand. Lime and cement were used in the proportions of about 2:1. The mortar was worked in well, so that there were no appreciable leaks. There was an air space between the brick and the flue liner that averaged about ½ in. across. The tile was laid one above the other without a mortar seal.

To close the chimney above the test height a sheet of metal reaching through at the top of the vents was wedged tightly across the chimney. To reduce the turbulence that would result if the flue gases were allowed to impinge against this flat surface, a sheet metal splitter was placed at the outlet as shown by the diagram in Fig. 1. Fig. 3 is a photograph of the two experimental brick chimneys showing the vents at several levels and the scaffolding from which the necessary operations on the chimneys during the investigation could be made.

TEST EQUIPMENT

The heater used during the tests consisted of a radial gas burner somewhat less than 9-in. in diameter located in a 9-in. pipe. See Fig. 2. The burner was located about 18-in. above the floor. The pipe tapered above the burner to a diameter of 6 in. where it entered an elbow and the elbow was connected to the chimney by a horizontal 6-in. pipe, about 4 ft long. This pipe

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TABLE 1—TEMPERATURE AND DRAFT IN 9 IN. X 9 IN. MASONRY CHIMNEY[a]

CHIMNEY HEIGHT, FT		FUEL OIL EQUIVALENT OF CHIMNEY GASES		FLUE GAS TEMP. AT INLET, F	AVERAGE CHIMNEY GAS TEMP. F	OUTSIDE TEMP. F	OBSERVED DRAFT INCH, W. G.	COMPUTED STATIC DRAFT INCH, W. G.	CORRECTED DRAFT[b]		INDICATED FRICTION LOSS, INCH, W. G.
NOMINAL	HEIGHT ABOVE THIMBLE	FUEL OIL GAL/HR	CO_2 PER CENT						OBSERVED INCH, W. G.	COMPUTED STATIC INCH, W. G.	
35	31.5	0.5	10	236	142	68	0.040	0.053	0.076	0.089	+0.013
35	31.5	1.5	8	200	158	81	0.043	0.055	0.086	0.098	+0.012
35	31.5	0.5	10	400	224	77	0.081	0.095	0.128	0.142	+0.014
35	31.5	1.5	8	400	278	80	0.092	0.117	0.139	0.164	+0.025
35	31.5	0.5	10	1000	488	77	0.163	0.191	0.206	0.234	+0.028
35	31.5	1.5	8	1000	601	78	0.169	0.214	0.216	0.261	+0.045
30	27.5	0.5	10	250	160	83	0.043	0.049	0.070	0.076	+0.006
30	27.5	1.5	8	250	195	84	0.055	0.064	0.097	0.106	+0.009
30	27.5	0.5	10	400	213	68	0.075	0.084	0.107	0.116	+0.009
30	27.5	1.5	8	400	269	72	0.094	0.108	0.125	0.139	+0.014
30	27.5	0.5	10	1000	460	64	0.146	0.169	0.174	0.197	+0.023
30	27.5	1.5	8	1000	608	72	0.168	0.193	0.204	0.228	+0.025
25	22.3	0.5	10	250	171	92	0.034	0.037	0.072	0.076	+0.003
25	22.3	1.5	8	200	171	94	0.034	0.036	0.074	0.076	+0.002
25	22.3	0.5	10	400	243	78	0.066	0.073	0.097	0.105	+0.006
25	22.3	1.5	8	400	301	79	0.085	0.091	0.116	0.123	+0.006
25	22.3	0.5	10	1000	504	79	0.124	0.137	0.154	0.167	+0.013
25	22.3	1.5	8	1000	642	80	0.146	0.159	0.177	0.190	+0.013
20	17.2	0.5	10	250	179	94	0.029	0.032	0.059	0.062	+0.003
20	17.2	1.5	8	200	188	94	0.031	0.032	0.059	0.060	+0.001
20	17.2	0.5	10	400	270	81	0.058	0.062	0.082	0.086	+0.004
20	17.2	1.5	8	400	297	55	0.075	0.081	0.087	0.093	+0.006
20	17.2	0.5	10	600	330	50	0.086	0.091	0.095	0.100	+0.005
20	17.2	1.5	8	600	425	55	0.104	0.106	0.116	0.118	+0.002
20	17.2	0.5	10	1000	526	37	0.120	0.130	0.123	0.134	+0.010
20	17.2	1.5	8	915	588	44	0.129	0.134	0.135	0.141	+0.005
15	11.3	0.5	10	250	182	87	0.023	0.023	0.041	0.041	+0.000
15	11.3	1.5	8	200	180	84	0.025	0.024	0.042	0.041	−0.001
15	11.3	0.5	10	400	258	32	0.053	0.055	0.054	0.056	+0.002
15	11.3	1.5	8	400	293	43	0.056	0.057	0.060	0.061	+0.001
15	11.3	0.5	10	600	369	34	0.068	0.071	0.069	0.071	+0.003
15	11.3	1.5	8	600	428	40	0.072	0.076	0.075	0.078	+0.004
15	11.3	0.5	10	1000	577	32	0.094	0.092	0.094	0.092	−0.002
15	11.3	1.5	8	1000	654	55	0.092	0.090	0.100	0.098	−0.002

[a] Actual inside dimensions of flue lining 7¼ x 7¼ in.
[b] Corrected for Outside Temp. 32 F. Barometer 29.92 In. Hg.

TABLE 2—TEMPERATURE AT CENTER OF FLUE GAS STREAM IN 9 IN. X 9 IN. MASONRY CHIMNEY[a]

CHIMNEY HEIGHT, FT		FUEL OIL EQUIVALENT OF CHIMNEY GASES		FLUE GAS TEMP. AT INLET, F	THERMOCOUPLE STATION NO.						
NOMINAL	HEIGHT ABOVE THIMBLE	FUEL OIL GAL/HR	CO_2 PER CENT		1	2	3	4	5	6	7
35	31.5	0.5	10	236	178	160	158	139	130	115	110
35	31.5	1.5	8	200	177	174	—	157	—	143	—
35	31.5	0.5	10	400	304	266	251	223	205	184	180
35	31.5	1.5	8	400	330	317	302	275	257	238	228
35	31.5	0.5	10	1000	666	550	510	448	400	346	325
35	31.5	1.5	8	1000	767	723	681	607	554	501	471
30	27.5	0.5	10	200	189	175	169	152	145	130	.....
30	27.5	1.5	8	250	213	211	206	190	182	173	.....
30	27.5	0.5	10	400	289	243	229	200	174	141	.....
30	27.5	1.5	8	400	314	312	298	272	243	237	.....
30	27.5	0.5	10	1000	637	542	510	427	358	276	.....
30	27.5	1.5	8	1000	722	701	664	585	497	479	.....
25	22.3	0.5	10	250	199	180	173	153	148	.....	.....
25	22.3	1.5	8	200	180	177	174	152	136	.....	.....
25	22.3	0.5	10	400	300	275	240	208	190	.....	.....
25	22.3	1.5	8	400	341	321	300	276	260	.....	.....
25	22.3	0.5	10	1000	641	536	514	426	382	.....	.....
25	22.3	1.5	8	1000	735	701	664	580	528	.....	.....
20	17.2	0.5	10	250	199	181	177	158	.....	.....	.....
20	17.2	1.5	8	200	179	177	175	164	.....	.....	.....
20	17.2	0.5	10	400	296	249	241	219	.....	.....	.....
20	17.2	1.5	8	400	319	309	293	274	.....	.....	.....
20	17.2	0.5	10	900	303	334	314	282	.....	.....	.....
20	17.2	1.5	8	900	438	427	410	367	.....	.....	.....
20	17.2	0.5	10	1000	522	508	480	425	.....	.....	.....
20	17.2	1.5	8	915	675	617	598	556	.....	.....	.....
15	11.3	0.5	10	250	200	177	170	.....	.....	.....	.....
15	11.3	1.5	8	200	182	180	177	.....	.....	.....	.....
15	11.3	0.5	10	400	272	263	240	.....	.....	.....	.....
15	11.3	1.5	8	400	300	300	279	.....	.....	.....	.....
15	11.3	0.5	10	600	383	357	336	.....	.....	.....	.....
15	11.3	1.5	8	600	443	440	402	.....	.....	.....	.....
15	11.3	0.5	10	1000	596	602	532	.....	.....	.....	.....
15	11.3	1.5	8	1000	680	672	611	.....	.....	.....	.....

[a] Actual inside dimensions of flue lining 7½ x 7½ in.

and the thimble of the chimney were about 6 ft above the floor. The heater and smoke pipe were insulated with 4-in. of rock wool, covered with metal foil.

The draft was measured by means of a copper tube which entered the smoke pipe about 18 in. from the nearest inside surface of the chimney. The axis of the tube was normal to the flow of gases. The tube was connected to a draft gage of the inclined type, in which methyl alcohol was used as the indicating fluid and which was graduated in thousandths of an inch of water

TABLE 3—TEMPERATURE OF INNER FLUE LINING SURFACE IN 9 IN. X 9 IN. MASONRY CHIMNEY[a]

CHIMNEY HEIGHT, FT		FUEL OIL EQUIVALENT OF CHIMNEY GASES		FLUE GAS TEMP. AT INLET, F	THERMOCOUPLE STATION NO.						
NOMINAL	HEIGHT ABOVE THIMBLE	FUEL OIL GAL/HR.	CO_2 PER CENT		1	2	3	4	5	6	7
35	31.5	0.5	10	236	175	144	158	127	108	102	95
35	31.5	1.5	8	200	172	156	—	136	—	122	—
35	31.5	0.5	10	400	291	248	240	193	160	139	135
35	31.5	1.5	8	400	320	292	268	220	203	203	185
35	31.5	0.5	10	1000	654	530	510	373	290	286	271
35	31.5	1.5	8	1000	761	663	602	510	435	399	390
30	27.5	0.5	10	200	150	130	144	120	135	129	.....
30	27.5	1.5	8	250	189	192	174	172	162	167	.....
30	27.5	0.5	10	400	221	215	184	173	147	—	.....
30	27.5	1.5	8	400	276	272	250	237	206	204	.....
30	27.5	0.5	10	1000	519	475	390	357	282	162	.....
30	27.5	1.5	8	1000	625	611	510	493	418	402	.....
25	22.3	0.5	10	250	150	165	147	142	127	.....	.....
25	22.3	1.5	8	200	104	105	—	102	140	.....	.....
25	22.3	0.5	10	400	239	223	183	188	157	.....	.....
25	22.3	1.5	8	400	294	280	242	249	214	.....	.....
25	22.3	0.5	10	1000	536	475	369	367	300	.....	.....
25	22.3	1.5	8	1000	637	611	507	503	418	.....	.....
20	17.2	0.5	10	250	163	168	143	144	.....	.....	.....
20	17.2	1.5	8	200	104	105	148	152	.....	.....	.....
20	17.2	0.5	10	400	238	245	215	188	.....	.....	.....
20	17.2	1.5	8	400	300	278	249	260	.....	.....	.....
20	17.2	0.5	10	900	330	321	272	236	.....	.....	.....
20	17.2	1.5	8	900	382	390	351	320	.....	.....	.....
20	17.2	0.5	10	1000	543	497	417	330	.....	.....	.....
20	17.2	1.5	8	915	550	579	503	439	.....	.....	.....
15	11.3	0.5	10	250	161	166	139	.....	.....	.....	.....
15	11.3	1.5	8	200	167	168	151	.....	.....	.....	.....
15	11.3	0.5	10	400	205	190	193	.....	.....	.....	.....
15	11.3	1.5	8	400	256	242	232	.....	.....	.....	.....
15	11.3	0.5	10	600	333	308	273	.....	.....	.....	.....
15	11.3	1.5	8	600	383	361	336	.....	.....	.....	.....
15	11.3	0.5	10	1000	530	449	449	.....	.....	.....	.....
15	11.3	1.5	8	1000	594	581	523	.....	.....	.....	.....

[a] Actual inside dimensions of flue lining 7½ x 7½ in.

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TABLE 4—TEMPERATURE AND DRAFT IN 9 IN. X 13 IN. MASONRY CHIMNEY[a]

Chimney Height Ft.		Fuel Oil Equivalent of Chimney Gases		Flue Gas Temp. at Inlet, F	Average Chimney Gas Temp. F	Outside Temp. F	Observed Draft Inch W. G.	Computed Static Draft Inch W. G.	Corrected Draft[b]		Indicated Friction Loss, Inch W. G.
Nominal	Height Above Thimble	Fuel Oil Gal./Hr.	CO_2 Per Cent						Observed Inch W. G.	Computed Static Inch W. G.	
35	31.5	0.5	10	394	125	78	0.035	0.034	0.073	0.077	−0.001
35	31.5	1.5	8	566	153	82	0.047	0.032	0.093	0.098	+0.005
35	31.5	1.5	6	600	352	84	0.140	0.141	0.193	0.194	+0.001
35	31.5	0.5	10	600	372	76	0.104	0.117	0.192	0.165	+0.013
35	31.5	0.5	10	1000	428	77	0.163	0.178	0.205	0.221	+0.015
35	31.5	1.5	8	1000	542	84	0.189	0.201	0.230	0.247	+0.012
30	27.5	0.5	10	390	148	83	0.040	0.045	0.081	0.086	+0.005
30	27.5	1.5	8	552	189	83	0.048	0.047	0.089	0.088	−0.001
30	27.5	0.5	10	605	378	80	0.080	0.103	0.120	0.144	+0.014
30	27.5	1.5	8	585	379	72	0.130	0.144	0.161	0.175	+0.014
30	27.5	0.5	10	1007	435	81	0.155	0.148	0.179	0.191	+0.013
30	27.5	1.5	8	995	569	89	0.166	0.168	0.225	0.227	+0.002
25	22.5	0.5	10	236	155	86	0.034	0.034	0.069	0.069	0.000
25	22.5	1.5	8	390	186	86	0.039	0.037	0.074	0.072	−0.002
25	22.5	0.5	10	600	312	86	0.085	0.090	0.122	0.126	+0.004
25	22.5	1.5	8	592	362	83	0.108	0.110	0.149	0.144	+0.004
25	22.5	0.5	10	1007	464	85	0.120	0.127	0.154	0.181	+0.027
25	22.5	1.5	8	1007	607	84	0.151	0.153	0.184	0.186	+0.002
20	17.2	0.5	10	280	175	79	0.030	0.028	0.068	0.060	−0.002
20	17.2	1.5	8	330	192	81	0.034	0.041	0.080	0.076	−0.003
20	17.2	0.5	10	598	324	74	0.069	0.079	0.099	0.089	+0.010
20	17.2	1.5	8	604	410	74	0.090	0.094	0.111	0.115	+0.004
20	17.2	0.5	10	1015	495	79	0.094	0.107	0.115	0.128	+0.013
20	17.2	1.5	8	1000	645	74	0.122	0.127	0.140	0.147	+0.005
15	11.2	0.5	10	270	172	69	0.022	0.021	0.041	0.040	−0.001
15	11.2	1.5	8	300	186	65	0.022	0.020	0.040	0.036	−0.002
15	11.2	0.5	10	504	256	66	0.055	0.032	0.073	0.070	−0.003
15	11.2	1.5	8	504	336	66	0.056	0.062	0.077	0.081	+0.004
15	11.2	0.5	10	1000	500	82	0.065	0.074	0.082	0.061	+0.006
15	11.2	1.5	8	1000	692	65	0.083	0.082	0.101	0.100	−0.001

[a] Actual inside dimensions of flue lining 6½ x 11 in.
[b] Corrected for Outside Temp. 32 F. Barometer 29.92 In. Hg.

pressure. The gages were calibrated and checked occasionally against a Wahlen gage.

Temperature in the chimney and smoke pipe was observed with unshielded thermocouples made of 28 gage chromel and alumel wire. The potentiometer used was a semi-precision instrument which could easily be read to 1 F.

One pair of thermocouples was located in the smoke pipe about 18 in. from the nearest inside wall of the chimney. One of these was placed against the inner wall of the smoke pipe, the other in the center of the flue gas stream. Two thermocouples were inserted through holes in the side of the chimney

TABLE 5—TEMPERATURE AT CENTER OF FLUE GAS STREAM IN 9 IN. X 13 IN. MASONRY CHIMNEY[a]

Chimney Height, Ft		Fuel Oil Equivalent of Chimney Gases		Flue Gas Temp. at Inlet, F	Thermocouple Station No.						
Nominal	Height Above Thimble	Fuel Oil Gal./Hr.	CO_2 Per Cent		1	2	3	4	5	6	7
35	31.5	0.5	10	394	163	139	122	111	105	101	102
35	31.5	1.5	8	566	181	168	150	131	144	141	142
35	31.5	1.5	6	600	471	407	374	340	312	299	293
35	31.5	0.5	10	600	415	334	288	250	235	207	197
35	31.5	0.5	10	1000	675	528	459	403	359	323	307
35	31.5	1.5	8	1000	781	635	560	513	465	428	408
30	27.5	0.5	10	390	199	172	156	143	133	127	.....
30	27.5	1.5	8	552	180	169	153	152	147	144	.....
30	27.5	0.5	10	605	413	319	281	239	217	205	.....
30	27.5	1.5	8	585	481	413	384	345	333	333	.....
30	27.5	0.5	10	1007	694	521	455	401	361	335	.....
30	27.5	1.5	8	995	788	655	590	523	488	450	.....
25	22.5	0.5	10	358	196	164	150	138	1389	.....	.....
25	22.5	1.5	8	390	179	166	160	150	1439	.....	.....
25	22.5	0.5	10	600	418	325	305	290	2410	.....	.....
25	22.5	1.5	8	592	479	407	383	355	3049	.....	.....
25	22.5	0.5	10	1007	851	625	443	379	2468	.....	.....
25	22.5	1.5	8	1007	794	647	566	481	4705	.....	.....
20	17.2	0.5	10	280	215	179	169	1435	.....	.....	.....
20	17.2	1.5	6	330	215	195	184	1745	.....	.....	.....
20	17.2	0.5	10	598	434	343	290	2525	.....	.....	.....
20	17.2	1.5	8	604	488	418	383	3545	.....	.....	.....
20	17.2	0.5	10	1015	903	614	430	3726	.....	.....	.....
20	17.2	1.5	8	1000	791	661	566	4459	.....	.....	.....
15	11.2	0.5	10	270	187	169	1305	.....	.....	.....	.....
15	11.2	1.5	8	300	179	165	1425	.....	.....	.....	.....
15	11.2	0.5	10	504	439	341	2935	.....	.....	.....	.....
15	11.2	1.5	8	504	507	419	3485	.....	.....	.....	.....
15	11.2	0.5	10	1000	981	698	470	.....	.....	.....	.....
15	11.2	1.5	8	1000	955	663	500	.....	.....	.....	.....
15	11.2	0.5	10	1000	718	524	395	.....	.....	.....	.....
15	11.2	1.5	8	995	834	645	505	.....	.....	.....	.....

[a] Actual inside dimensions of flue lining 6½ x 11 in.
[b] Thermocouples not in place. Temperatures were estimated from readings of previous tests.

at each of seven points beginning about 18 in. above the flue gas inlet and continuing at intervals of about 5 ft to the top of the chimney. In order to hold these thermocouples rigidly in position, the wires except the junctions were encased in 3/16 in. porcelain insulators. The insulators of each pair were fastened together in such a way that when one of the thermocouples was pushed through the hole in the chimney until it touched the opposite wall, the other stopped at the center of the cross section. Another thermo-

FIG. 4. PHOTOGRAPH OF FLUE GAS PRODUCING EQUIPMENT WITH AIR METER SHOWN IN FOREGROUND

couple at each of these levels of the chimney was held against the outer brick surface.

In some of the tests the thermocouples were moved in steps away from the chimney wall and observations were made to find the change of temperature with distance between the flue wall and the center of the gas stream. In other tests draft tubes were used to measure draft at the various levels in the chimney.

A cross section of the flue gas producing equipment and the location of the draft tubes and thermocouples stations are shown on Fig. 2. A photograph of the flue gas producing equipment is shown in Fig. 4.

It is obvious that the thermocouples with junctions placed in the axis of the chimney do not measure the average temperature of the gases, and that

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TABLE 6—COMPARISON OF PERFORMANCE OF 9 IN. X 9 IN. AND 9 IN. X 13 IN. MASONRY CHIMNEYS[a]

CHIMNEY HEIGHT, FT		FUEL OIL EQUIVALENT OF CHIMNEY GASES		FLUE GAS TEMP. AT INLET, F	OBSERVED DRAFT CORRECTED[b] INCH W. G.		AVERAGE CHIMNEY GAS TEMP. F		TEMP. DROP AT CHIMNEY ENTRANCE F		TEMP. DROP IN CHIMNEY F	
NOMINAL	HEIGHT ABOVE THIMBLE	FUEL OIL GAL/HR.	CO_2 PER CENT		9 IN. X 9 IN.	9 IN. X 13 IN.	9 IN. X 9 IN.	9 IN. X 13 IN.	9 IN. X 9 IN.	9 IN. X 13 IN.	9 IN. X 9 IN.	9 IN. X 13 IN.
35	31.5	0.5	10	250	0.076	...	142	...	58	...	68	...
35	31.5	0.5	10	200	...	0.078	...	125	...	41	...	61
35	31.5	1.5	8	200	0.086	0.093	158	155	23	25	...	39
35	31.5	0.5	10	400	0.128	...	224	...	96	...	124	...
35	31.5	1.5	8	400	0.139	...	278	...	70	...	102	...
35	31.5	0.5	10	600	...	0.152	...	272	...	185	...	188
35	31.5	1.5	8	600	...	0.193	...	355	...	139	...	218
35	31.5	0.5	10	1000	0.206	0.205	488	438	331	325	344	368
35	31.5	1.5	8	1000	0.216	0.236	601	542	233	239	296	355
30	27.5	0.5	10	250	0.070	0.081	160	155	11	61	59	72
30	27.5	1.5	8	200	...	0.089	...	159	...	22	...	36
30	27.5	1.5	8	250	0.097	...	195	...	37	...	41	...
30	27.5	0.5	10	400	0.107	...	213	...	111	...	148	...
30	27.5	1.5	8	400	0.125	...	269	...	86	...	77	...
30	27.5	0.5	10	600	...	0.129	...	278	...	192	...	213
30	27.5	1.5	8	600	...	0.161	...	378	...	114	...	176
30	27.5	0.5	10	1000	0.174	0.179	460	438	363	313	361	364
30	27.5	1.5	8	1000	0.204	0.225	608	609	278	207	243	338
25	22.3	0.5	10	250	0.072	0.069	171	155	51	62	51	68
25	22.3	1.5	8	200	0.074	0.074	171	160	20	21	22	36
25	22.3	0.5	10	400	0.097	...	243	...	100	...	110	...
25	22.3	1.5	8	400	0.116	...	301	...	59	...	81	...
25	22.3	0.5	10	600	...	0.122	...	312	...	184	...	171
25	22.3	1.5	8	600	...	0.140	...	382	...	113	...	174
25	22.3	0.5	10	1000	0.154	0.154	504	462	359	356	259	311
25	22.3	1.5	8	1000	0.177	0.184	642	607	265	213	207	324
20	17.2	0.5	10	250	0.059	0.062	179	175	51	65	41	70
20	17.2	1.5	8	250	...	0.069	...	192	...	35	...	41
20	17.2	1.5	8	200	0.059	...	188	...	21	...	15	...
20	17.2	0.5	10	400	0.082	...	270	...	110	...	71	...
20	17.2	1.5	8	400	0.087	...	297	...	81	...	45	...
20	17.2	0.5	10	600	0.095	0.089	330	324	207	174	111	171
20	17.2	1.5	8	600	0.116	0.111	425	410	142	118	71	136
20	17.2	0.5	10	1000	0.123	0.128	526	496	368	350	203	295
20	17.2	1.5	8	1000	...	0.147	...	645	...	219	...	251
20	17.2	1.5	8	915	0.135	...	588	...	240	...	119	...
15	11.3	0.5	10	250	0.041	0.041	182	172	50	83	30	27
15	11.3	1.5	8	200	0.042	0.040	180	166	18	21	5	24
15	11.3	0.5	10	400	0.054	...	258	...	128	...	32	...
15	11.3	1.5	8	400	0.060	...	293	...	100	...	21	...
15	11.3	0.5	10	600	0.075	0.073	369	356	218	171	44	138
15	11.3	1.5	8	600	0.069	0.082	428	431	157	92	41	139
15	11.3	0.5	10	1000	0.094	0.082	577	560	402	324	66	211
15	11.3	1.5	8	1000	0.100	0.101	654	682	320	271	69	253

[a] Data assembled from Tables 1–5 inclusive.

[b] Corrected to Outside Temp. 32 F, Barometer 29.92 In. Hg.

they are affected by radiation to the flue walls. However for the purpose of this investigation it appeared sufficiently accurate to consider the temperatures indicated by the thermocouples as representative of average gas temperatures, for such purposes as computing static draft, etc.

In the following discussion the chimneys are referred to as *15 ft chimneys, 20 ft chimneys,* etc. These heights refer to the approximate height of the chimney above the burner level. The actual heights of the chimneys were somewhat more than these nominal heights. The computed drafts given in the discussion and the tables are however based on the effective height of the chimney above the level of the draft measuring tube or center of the thimble. The relations between the effective and nominal chimney heights were:

NOMINAL HEIGHT, FEET	15	20	25	30	35
Height above burner level, feet.....	15.6	21.5	26.6	31.8	35.8
Height above draft tube, feet.....	11.3	17.2	22.3	27.5	31.5

GENERAL OBSERVATIONS

The results of the tests of the brick chimneys are shown in Tables 1 to 7 inclusive in this report.

It was observed that the temperature on the inner surface of the flue lining was in most cases somewhat lower than that in the center of the flue gas stream at the same level. There was a rapid change in temperature with horizontal distance in the layer of gases next to the wall of the flue, but except for this layer, an inch or less in thickness, the temperature across the chimney was found to approach uniformity.

In one test the temperature of the wall of the flue was 255 F, at the center of the gas stream the temperature was 367 F, while at other points one inch or more away from the flue wall temperatures varying from 351 F to 383 F were observed. In another test the flue wall and center temperatures were, respectively, 215 F and 355 F, while temperatures at other points varied from 349 F to 367 F. Other tests showed a comparatively small range of temperatures across the chimney, except in the layers near the flue wall. In general the temperature at the center was somewhat above the average, but considering the greater velocity of the gases at the center it was concluded that for practical purposes the temperature at the center of the gas stream could be accepted as representative of that of the flue gases at the same level. Accordingly, the temperatures measured at the center of the chimney were used in the computations of static draft.

THE FIFTEEN FOOT CHIMNEY

When measured at the inlet to the chimney, the available draft in the 15 ft chimney changed from about 0.054 in. W. G. with the entering gases at 400 F, to about 0.094 in. W. G. with the entering gases at 1000 F. Very little difference in the available draft was observed when the volume of flue gases was increased in the range of gas flow rates used. When the inlet gas tem-

TABLE 7—LOSS OF DRAFT BY FRICTION IN

FUEL OIL EQUIVALENT OF CHIMNEY GASES		FLUE GAS TEMP. AT INLET, F	OBSERVED DRAFT INCH, W. G.		OBSERVED DRAFT[a] DIFFERENCE INCH, W. G. STATION 1–7
FUEL OIL GAL/HR.	CO_2 PER CENT		STATION 1[a]	STATION 7	
0.5	10	200	0.043	0.007	0.036
1.5	8	200	0.041	0.004	0.037
0.5	10	400	0.073	0.008	0.065
0.5	10	1000	0.151	0.012	0.139
1.5	8	1000	0.174	0.024	0.150

[a] Vertical distance between stations 1 and 7 was 29 ft − 4 in.

perature was constant an increase in the temperature of the flue gases in the chimney was noted as the velocity of the gases was increased. This action is illustrated by data taken from the tables and presented here for convenience:

Entering Temp.	Deg Fahr	400	400	1000	1000
Flue Gas Flow	Cfm (Std.)	18.4	67.5	18.4	67.5
Gas Temp., bottom of chimney (Sta. 1)	Deg Fahr	272	300	598	680
Gas Temp., top of chimney (Sta. 3)	Deg Fahr	240	279	532	611

The curves in Fig. 5 and Table 1 show the available draft in the 15 ft chimney over a range of inlet temperatures from 400 to 1000 F for the combustion of fuel oil at various rates after continuous operation had proceeded long enough to reach steady temperature conditions. Fig. 5 also shows the equilibrium temperatures at two stations in the chimney for the various test conditions.

In each test, as noted previously, the gases entering the chimney were kept as nearly as possible at some constant temperature: 200, 400, 600, 800, or 1000 F. In the first few minutes of each test the draft increased very rapidly, usually coming nearly to a steady condition within half an hour. The temperatures at the chimney walls, however, continued to rise for about two hours. Figs. 6 and 7 show how the draft and temperature at the chimney entrance and the temperatures at the three stations in the 15-ft chimney increased with time after starting cold.

As the velocity of the flue gases in the chimney was increased, the temperature on the inner surface of the flue lining increased as well as the flue gas temperature at the center of the flue. When the flue gases entered the chimney at 400 F the surface temperature at a point 18 in. above the inlet was 205 F with 18.4 cu ft of gas flowing per minute and 258 F with 67.5 cfm. With flue gases entering at 1000 F the temperature at this point increased from 536 to 594 F, when the rate of flow was increased from 18.4 to 67.5 cfm. With a corresponding increase in velocity of flue gas the surface temperature at

9 IN. X 9 IN. MASONRY CHIMNEY

COMPUTED STATIC DRAFT INCH, W. G.		STATIC DRAFT DIFFERENCE INCH, W. G. STATION 1-7	DRAFT LOSS INCH W. G.	PER CENT OF COMPUTED STATIC DRAFT LOST BY FRICTION
STATION 1	STATION 7			
0.045	0.001	0.044	0.008	18
0.052	0.000	0.052	0.015	29
0.088	0.001	0.087	0.014	16
0.176	0.003	0.173	0.034	20
0.208	0.005	0.203	0.053	26

station 3 near the top of the chimney increased from 193 to 232 F with 400 F inlet temperature, and from 449 to 525 F with 1000 F inlet temperature.

In addition to the draft observations made at the inlet to the chimney others were made in the air duct between the large meter and the heating unit. With low rates of air delivery, 18.4 to 25 cfm, the draft at this point was slightly greater than that observed near the chimney inlet. As the volume of air supplied was increased the pressure became greater until, with a flow of 65 cfm a small positive pressure was noted. Since a positive pressure near the inlet to the heater indicates that the air is being forced into the unit, it appears that this volume exceeds the capacity of the chimney when attached to the particular heater used.

THE TWENTY FOOT CHIMNEY

During the tests of the 20 ft chimney changes in the weather permitted observation of the effect of differences in outside temperature on chimney draft. Whereas nearly all tests of the 15 ft chimney were made with the outside temperature in the range from 30 to 40 F, tests of the 20 ft chimney were obtained at 28 F, at 81 F and at intermediate temperatures. A high outside temperature lowers the density of the air and since the available draft is proportional to the difference in density between chimney gases and air a lower draft results. For the sake of uniformity the values for the observed draft were corrected for an outside temperature of 32 F before plotting. A smooth curve was then obtained when the draft was plotted against the average chimney gas temperature. Corrections were also made for variation in barometric pressure. When these corrections had been made, the available draft at the standard condition was shown to change from about 0.082 in. W. G. for an inlet temperature of 400 F to about 0.135 for an inlet temperature of 1000 F.

It was observed with the 20 ft chimney height, as with some others, that the mean temperature in the chimney increased when the gas flow was increased while the temperature of the gases entering the chimney was maintained constant. The increase in mean chimney gas temperature was sufficient to cause an increase in available draft despite the greater friction loss to be expected from the greater mass flow. This effect is illustrated by a tabulation of values taken from the data:

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TEMP. GAS ENTERING CHIMNEY	DEG FAHR	400		600		800		1000	
Flue Gas Flow.....	Cfm (Std.)	18.4	67.5	18.4	67.5	18.4	67.5	18.4	55.2
Temp. in Chimney, Avg....	Deg Fahr	270	297	330	425	422	555	526	645
Obs. Draft.....	In. W. G.	0.058	0.075	0.086	0.104	0.106	0.119	0.120	0.132

The average temperatures of the chimney gases shown in Table 1 are the averages of the temperatures observed at the center of the flue gas stream at Stations 1, 2, 3 and 4 shown in Fig. 1. This average temperature was used in obtaining the computed values of static draft shown in the tables.

The difference between the computed static draft, described previously, and the observed draft is an indication of the friction and inertia effects in the chimney. The results of these computations indicate that the friction loss in the chimney was not great. The difference between the computed static draft and the observed draft was less than 0.01 in. W. G. No consistent increase in friction loss was indicated in the 20 ft chimney with increased air velocities in the range of the tests.

The draft loss in the heating equipment was considerable and increased as the flow of flue gases became greater. With 18.4 cfm of flue gases passing, the draft measured 1 ft above the burner was slightly greater than that measured at the inlet to the chimney. When the rate of flow was increased to 67.5 cfm the draft at this point was about 40 per cent less than that at the inlet to the chimney.

Tables 2 and 3 contain a summary of the temperatures observed at a steady state of operation at the various stations at the center of the flue gas stream, and on the inside surface of the flue liner.

25, 30 AND 35 FT CHIMNEYS

In view of the fact that the results of the tests of the 15 and 20 ft chimneys showed the same general characteristics, it was not believed necessary to conduct the complete series of tests for the higher chimneys. Accordingly, only the highest and lowest velocities of flue gases were used, and in the case of the 9 by 9 in. chimney the tests with 600 and 800 F entering gas temperatures were omitted. With the 9 by 13 in. chimney tests with 400 F and 800 F entering gas temperatures were omitted. The higher chimneys showed the same general characteristics as the 15 and 20 ft chimneys. Some representative data on the 9 by 9 in. chimney are presented herewith for illustration.

(1) With a constant inlet temperature, a higher average chimney gas temperature resulted when the velocity of the flue gases was increased. With gases entering the 25 ft chimney at 1000 F the average flue gas temperature increased from 504 F to 642 F, when the flow of flue gases was increased

from 18.4 cfm to 67.5 cfm. In the 30 ft chimney with the same change in conditions the average temperature of the flue gases increased from 460 F to 608 F. In the 35 ft chimney, the average gas temperature increased from 488 F to 601 F.

(2) With a constant inlet temperature the draft increased when the velocity was increased, due to the higher mean chimney gas temperature. In changing from 18.4 cfm to 67.5 cfm the corrected observed draft in the 25 ft chimney, with 1000 F inlet temperature, increased from 0.154 in. W. G. to 0.177 in.

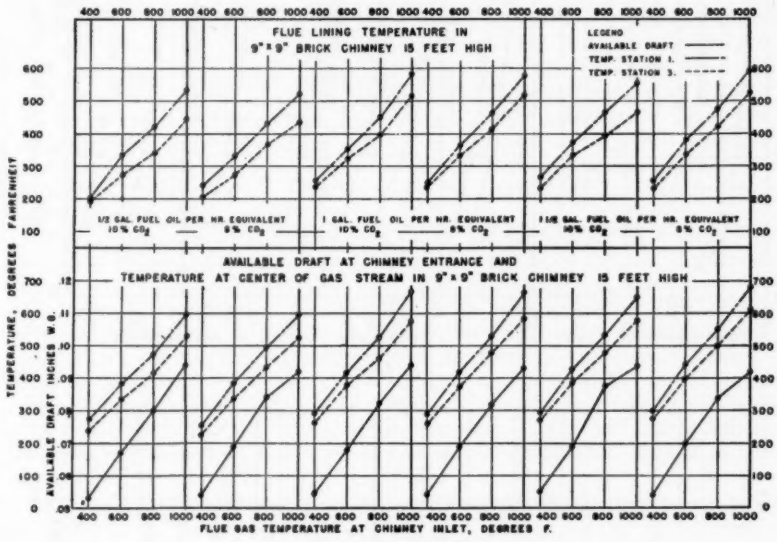
FLUE LINING TEMPERATURE IN
9" x 9" BRICK CHIMNEY 15 FEET HIGH
LEGEND
AVAILABLE DRAFT
TEMP. STATION 1.
TEMP. STATION 3.
1/2 GAL. FUEL OIL PER HR. EQUIVALENT
10% CO2 — 8% CO2
1 GAL. FUEL OIL PER HR. EQUIVALENT
10% CO2 — 8% CO2
1 1/2 GAL. FUEL OIL PER HR. EQUIVALENT
10% CO2 — 8% CO2
AVAILABLE DRAFT AT CHIMNEY ENTRANCE AND
TEMPERATURE AT CENTER OF GAS STREAM IN 9" x 9" BRICK CHIMNEY 15 FEET HIGH
TEMPERATURE, DEGREES FAHRENHEIT
AVAILABLE DRAFT INCHES W.G.
FLUE GAS TEMPERATURE AT CHIMNEY INLET, DEGREES F.

FIG. 5. CHIMNEY TEMPERATURES AND DRAFT FOR VARIOUS FLUE GAS TEMPERATURES

W. G. In the 30 ft chimney the draft increased from 0.174 in. W. G. to 0.204 in. W. G.; in the 35 ft chimney it increased from 0.206 to 0.216 in. W. G.

(3) In the 15 and 20 ft chimneys, the change in the indicated friction loss as the gas temperature was increased was smaller than the uncertainty in the data, but with the 25 ft and higher chimneys, there was a definite indication of such increase.

(4) In the tests of the 25 ft chimney as in the lower chimneys the draft loss due to friction was not much larger for the higher flue gas velocities used than for the lower velocities used during the tests. With the 30 ft and 35 ft chimney a measurable increase in the draft loss due to friction was noted for the higher flue gas velocities.

The observed draft in the upper portions of the higher chimneys was found to be greater than the computed static draft. In one case five feet from

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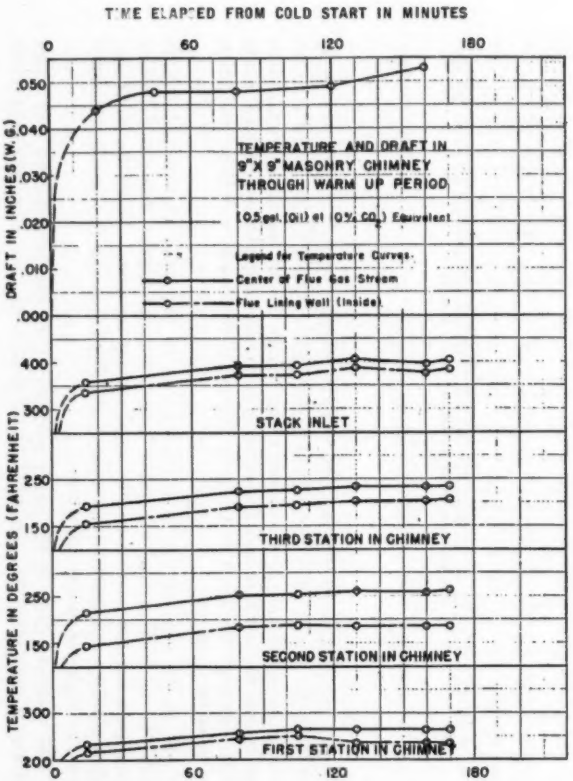

FIG. 6. TEMPERATURE AND DRAFT IN 15 FT 9 IN. X 9 IN. MASONRY CHIMNEY THROUGH WARM UP PERIOD

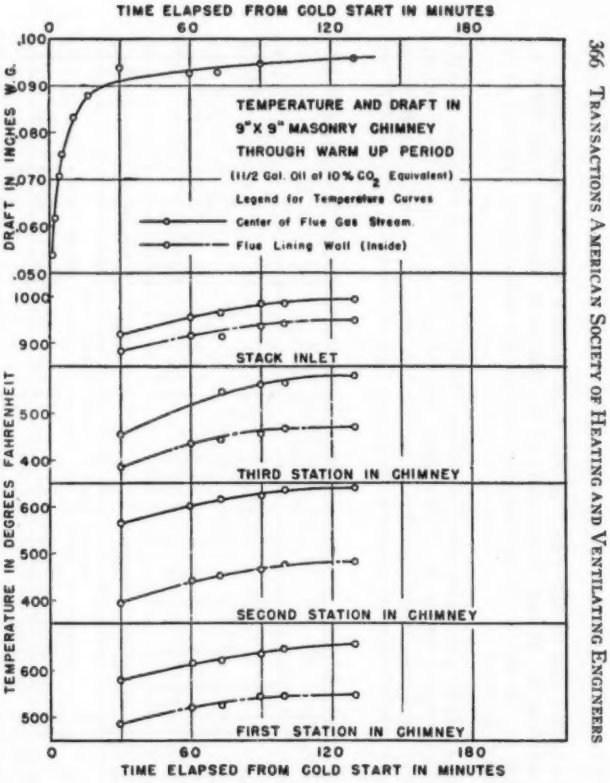

FIG. 7. TEMPERATURE AND DRAFT IN 9 IN. X 9 IN. MASONRY CHIMNEY THROUGH WARM UP PERIOD

the top, the draft observed was 0.005 in. W. G. more than the computed draft at the same level and 8 in. below the top it was 0.007 in. W. G. more. In other tests at different temperatures the draft observed 8 in. below the top of the chimney exceeded the computed draft by 0.005 to 0.02 in. W. G. This seemed to indicate a chimney effect above the actual top of the chimney. Some measurements were made which indicated the existence of draft more than 2 ft above the top of the chimney. The existence of the *extra* chimney effect of the column of hot gases over the chimney complicates calculation of the draft lost by friction. However, some general conclusions may be reached by comparison of the computed and observed drafts at different levels in the chimney. If the draft observed at one point is subtracted from that observed at some lower point the result is the draft produced between those points. The draft can be calculated and a figure for draft loss in a section of the chimney can be obtained. In this way the draft loss between Stations 1 and 7 in the chimney was calculated for five tests. The maximum friction loss indicated in this way was 29 per cent of the computed static draft. This loss was obtained with a flow of 67.5 cfm with flue gases entering the chimney at 200 F. The losses of draft for other tests determined in this way are given in Table 7.

This investigation was intended to supply experimental data sufficient to answer a definite question, namely; if a known quantity of flue gas at a known temperature flows into a particular chimney, what will the chimney draft be? Two sizes of chimneys, a number of chimney heights, several rates of gas flow and several temperatures at the inlet to the chimney, were used, covering most of the combinations of conditions for which an answer might be desired. The essential measurements were therefore those of rate of gas flow and gas temperature and draft at the chimney inlet. Numerous other measurements of draft and temperature at other points were made, but these are distinctly subsidiary though they may prove useful if a more detailed analysis of the data should be attempted.

DISCUSSION

H. F. RANDOLPH, Utica, N. Y.: I would like to inquire if any comparative data are available for lined and unlined chimneys?

MR. DILL: Not at the present time, but as far as friction is concerned, I believe the statements reported in the paper will hold, because the chimney velocities are not great enough to have any bearing upon the roughness of a masonry chimney. Of course, if a chimney were a quarter or half constricted, there would be a measurable effect.

QUESTION: Was any attempt made to determine the amount of leakage into the chimney?

MR. DILL: No leakage determinations were made, but the chimney was built by a contractor familiar with bricklaying and it was assumed that this chimney was a typical job. The lining tile were placed together with mortar, but they were not flushed.

R. K. THULMAN, Washington, D. C.: May we conclude from this paper that, in order to obtain a draft sufficient to operate, for example, a space heater requiring 0.06 in. draft, a masonry chimney of 15 ft is inadequate?

MR. DILL: The answer to this question depends a great deal on the operation of the unit and whether the chimney stays warm. If a space heater is connected to the chimney with a pot type vaporizing oil burner and it is operated for a period of less than a half an hour, the probabilities are there will be a period of smoking, because the device will operate with a draft that is too low for its proper functioning. Some of these units operate with 0.06 in. of draft and others demand 0.08 in., so under such circumstances the brick chimney has the disadvantage of great heat capacity, which requires some time for the proper chimney temperature to be attained.

MR. THULMAN: May I add to my previous question the assumption that the flue gases entering the chimney from the space heater are less than 400 F?

MR. DILL: In this case I do not believe sufficient draft would be available and under such circumstances a blower on the burner would be desirable.

C. E. OLSEN, Elyria, O.: It seems to me that these tests have been made under rather unfavorable conditions with the four sides of the chimney entirely exposed. In practice, an outside chimney generally has at least one side facing an inside temperature of 70 F.

MR. DILL: I do not believe that this was a serious matter, for the reason that the draft inside the chimney and the temperature on the inside of the walls came to equilibrium in about half an hour, whereas, the outside temperature never approached equilibrium.

QUESTION: It seems to me that this testing arrangement placed the chimney and the fan in series and I wonder how you compensated for that? By increasing the speed of the fan, did you not produce a pressure at the base of the chimney?

MR. DILL: Insofar as the chimney is concerned it does not know whether there is a motor, a fan, a meter or simply a damper at the point where the gases enter. As long as the desired draft is produced at a point approximating the place where the hot gases are liberated from the heating unit, the desired performance conditions will be maintained.

MR. THULMAN: Will you please make a few remarks about the flue gas temperature in long smoke pipe connections.

MR. DILL: I recall an installation of a space heater connected to a chimney burning about a gallon of oil an hour. The temperature of the flue gases leaving the heater was 1,050 F. When the gases entered the chimney after passing through two lengths of stove pipe and an elbow, the flue gas temperature had dropped to 575 F. This brings up the question as to whether the space heater should be credited with the heating effect produced from the smoke pipe.

ALLEN JOHNSON, Primos, Pa.: Have you any comparisons between vitreous enameled stacks, metal stacks and brick stacks, insofar as the amount of draft each is capable of producing?

MR. DILL: Some data are available on this subject. The amount of draft available from a metal stack depends entirely on how well it is insulated. There is a large drop in temperature of the gases upon entering such a stack, because its heat capacity is low. This situation might be an advantage in starting a fire with an oil-burning device without a forced-draft fan.

No. 1213

HEAT LOSS THROUGH BASEMENT WALLS AND FLOORS

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Pittsburgh, Pa.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in its Research Laboratory at the Pittsburgh Experiment Station of the U. S. Bureau of Mines.

HEAT loss from warmed buildings has been the subject of intensive study which has in general placed heat loss estimates for design purposes on a scientific basis. Some phases of the problem, including heat loss from attics and basements, however, still require further study. The recommended methods for estimating these losses have received little attention, and as a result they are still based on what are practically *rule of the thumb* methods. As an example, the accepted method for estimating heat losses from spaces below the ground level is to assume the transmittance coefficient for the wall or floor in question, including the inside film conductance, and an estimated earth temperature of 45 F next to the outside surface of the wall.

The increased use of basement space for commercial purposes and living or club quarters has made it desirable that satisfactory design data for estimating heating requirements be made available. These data are not only required for the purpose of estimating the heating requirements for maintaining comfort, but also to avoid condensation. In many instances, beautifully designed and decorated basement living quarters have been found unsatisfactory because condensation has spoiled the decorative efforts or flooded the floor. These difficulties arise from the uncorrected practice of building these basement walls of concrete or stone masonry without consideration for their heat transmitting qualities. The correction of this condition must be found in the building of better insulated walls.

The ASHVE Technical Advisory Committee[1] on Heat Requirements of Buildings outlined a program for an initial study of certain aspects of this

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Presented at the 48th Annual Meeting of the American Society of Heating and Ventilating Engineers, Philadelphia, Pa., January, 1942.

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problem which included the building of a basement at the ASHVE Research Laboratory, incorporating the necessary laboratory facilities for making the desired observations. The results of the study, carried on during the 1940-41 heating season and the following summer, are contained in this report.

TEST SET-UP

A plan and a vertical section through the basement,—which as indicated includes no superstructure, are given in Figs. 1 and 2. The ceiling and roof, which consisted of 1 in. of structural insulation board, 2-in. x 10-in. joists, 1-in. planking and water-proofing, were covered with 18 in. of earth to reduce ceiling losses.

Since heat losses must necessarily be greatly influenced by earth temperatures near the side walls and floor of the basement, a series of thermocouples were sunk to depths of 6 ft or more below the earth surface and basement

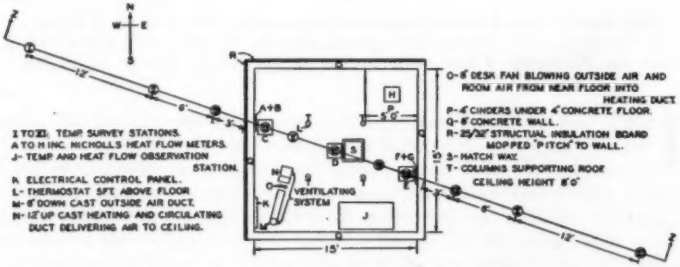
I TO XI. TEMP SURVEY STATIONS.
A TO H INC. NICHOLLS HEAT FLOW METERS.
J- TEMP AND HEAT FLOW OBSERVATION STATION.
K ELECTRICAL CONTROL PANEL.
L- THERMOSTAT 5FT. ABOVE FLOOR
M- 6" DOWN CAST OUTSIDE AIR DUCT.
N- 12" UP CAST HEATING AND CIRCULATING DUCT DELIVERING AIR TO CEILING.
O- 6' DESK FAN BLOWING OUTSIDE AIR AND ROOM AIR FROM NEAR FLOOR INTO HEATING DUCT
P- 4" CINDERS UNDER 4" CONCRETE FLOOR.
Q- 8" CONCRETE WALL.
R- 25/32" STRUCTUAL INSULATION BOARD MOPPED "PITCH" TO WALL.
S- HATCH WAY.
T- COLUMNS SUPPORTING ROOF
CEILING HEIGHT 8'0"
VENTILATING SYSTEM
15'
12'
6'

FIG. 1. PLAN VIEW OF BASEMENT AND SURROUNDING TEMPERATURE SURVEY STATIONS

floor at survey stations I to XI, along the line Z—Z (Fig. 1). At each of the survey stations a number of thermocouples were so placed as to give the temperature gradient from 6 in. above the ground or floor surface downward through the earth. The ground surface couples and those in the earth were in $\frac{3}{16}$-in. O.D. copper tubes so placed that 15 in. next to the junctions were in the isothermal plane. For the couples below the earth's surface this was accompanied by coiling the tubes containing the wire. The location and depth of the couples for each survey station are given in Fig. 2.

The side walls of the basement were 8-in., 1:2:4 mix concrete. The north and west walls were insulated with $\frac{25}{32}$-in. structural insulation board, attached to the outside of the concrete walls by mopping with pitch. The east and south walls were not insulated. The floor consisted of 4 in. of concrete placed directly on the earth, with the exception of a 5 ft x 5 ft section in the northeast corner, which was laid over 4 in. of cinders.

Fig. 3 shows the observation station equipped with a precision potentiometer, selective plug-in switches by which any of the 160 thermo-electric circuits, including those in the Nicholls' heat flow meters, could be read; while Fig. 4 shows another corner of the basement including two of the 2-ft square meters, *A* and *B*, attached respectively, with their centers 2 ft from the ceiling and 2 ft from the floor on the insulated side wall, and a third

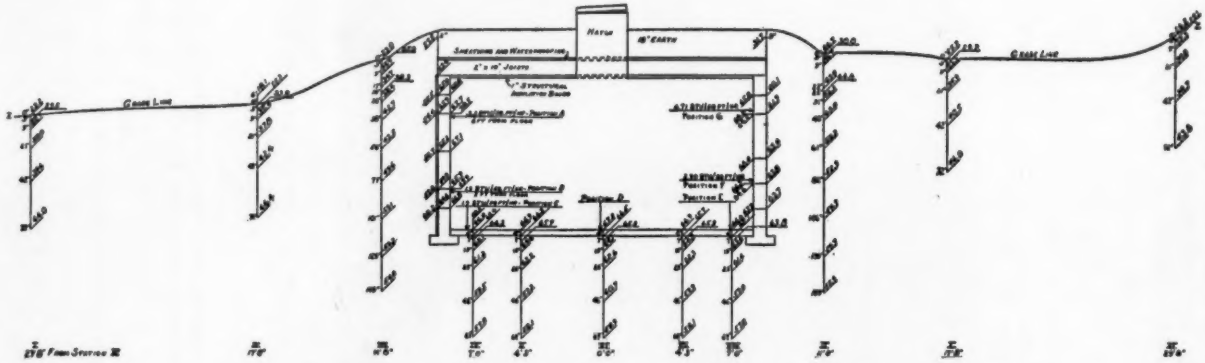

FIG. 2. TEMPERATURE AND HEAT FLOW PATTERN, SECTION ZZ, FIG. 1, FOR FEBRUARY 11, 1941, SHOWING TEMPERATURE SURVEY STATIONS I TO XI, DISTANCES BELOW EARTH OR BASEMENT FLOOR SURFACE AND OBSERVED TEMPERATURES; ALSO NICHOLLS' HEAT FLOW METER POSITIONS *A* TO *G* AND HEAT FLOW OBSERVATIONS

meter, *C*, attached to the floor. The thermostat, with its sensitive element at the 5-ft level, is also shown.

The basement space was heated electrically by thermostatically-controlled strip heaters placed in a 12-in. square vertical duct taking air from near the floor and discharging it near the ceiling. The space was ventilated by a downcast, 6-in. duct, leading from the outside down to the floor through an elbow, discharging the cold outside air at a point where it would be *picked up* by the intake of the circulating duct containing the heaters. A desk fan between

FIG. 3. INTERIOR OF BASEMENT SHOWING SET-UP FOR OBSERVING TEMPERATURES AND HEAT FLOW THROUGH WALLS AND FLOOR

the discharge from the vent duct and the intake to the circulating and heating duct served to increase the air flow from the floor to ceiling, so as to give approximately a 10-min. recirculation rate for the basement. The outside air supply was controlled by damper to give about 10 cfm. Exhaust air was permitted to escape through the door of the hatch, which was the only means of ingress and egress.

Tests included the observation of the temperatures at all of the points indicated in Fig. 1, including the outside and inside surfaces of the walls and

floor, between the insulation and concrete where insulation was applied, and the air 6 in. from the inside of the wall and floor at all points where meters were located.

TEST RESULTS

Fifty-four tests were made, first twice weekly from December 23, 1940, to May 1, 1941, and later at less frequent intervals until December 4, 1941.

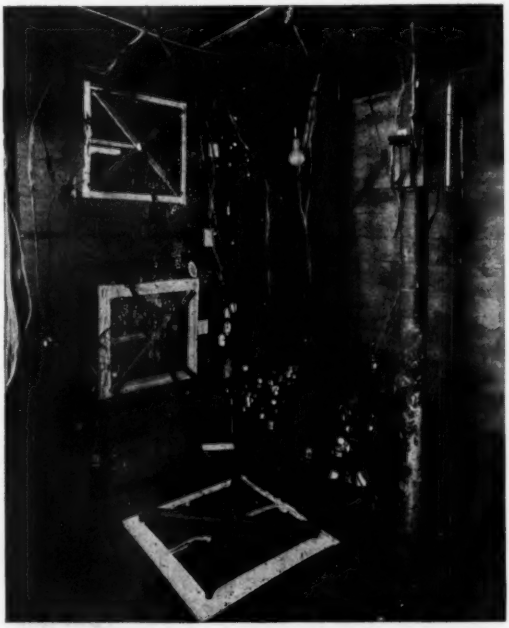

FIG. 4. NORTHWEST CORNER OF BASEMENT SHOWING HEAT FLOW METERS IN PLACE AT POSITIONS *A* AND *B* ON THE WEST WALL AND POSITION *C* ON THE FLOOR. THERMOSTAT IS ALSO SHOWN

Fig. 5 gives a log of Weather Bureau high and low temperature observations for each day and those observed for each test for points 9, 21, 42 and 72 in. below the ground surface at station I, 21 ft from the outside of the west basement wall.

The following values in relation to the time of the year, for positions *A* and *B* on the west wall, 2 ft, respectively, from the ceiling and floor, are given in Fig. 6: (1) temperature of the air 6 in. inside of the wall, (2) the temperature of the earth 44 in. from the inside wall surface, and (3) the

FIG. 5. LOG OF WEATHER BUREAU MAXIMUM AND MINIMUM DAILY TEMPERATURES IN PITTSBURGH AND TEMPERATURES AS OBSERVED IN SURVEY STATION I (SEE FIG. 1) AT DEPTHS OF 9, 21, 42 AND 72 IN.

heat flow. Fig. 7 gives similar data for locations *F* and *G*, 2 ft from the floor and 2 ft from the ceiling on the east, uninsulated wall, and Fig. 8 gives same data for locations *C*, *D*, and *E* on the floor. This figure also gives the heat flow for the position *H* on the floor where the concrete was placed over the 4 in. of cinders.

The earth temperatures around the perimeter of the side walls and floor of the basement in a vertical plane are given in Fig. 9 as observed at survey station III from the surface of the earth down to a point 38 in. below the basement floor level, thence at points 46 in. below the floor level at survey

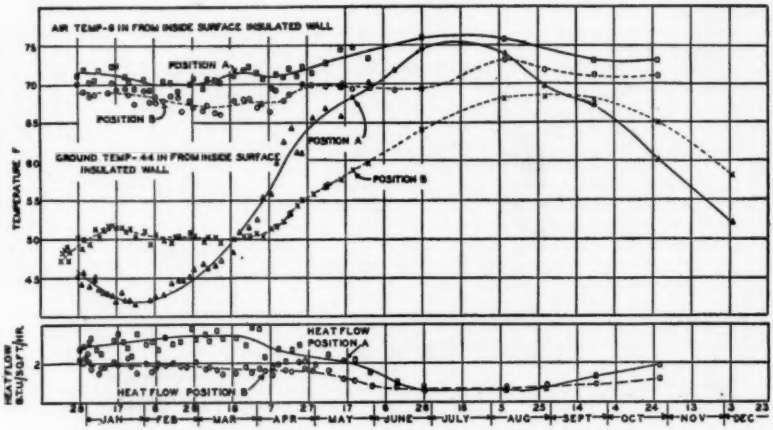

FIG. 6. RELATION BETWEEN TIME AND (1) AIR TEMPERATURE 6 IN. FROM INSIDE WALL SURFACE (2) GROUND TEMPERATURE 44 IN. FROM INSIDE WALL SURFACE, AND (3) HEAT FLOW FOR POSITIONS *A* AND *B* ON INSULATED WALL

stations IV, V, VI, VII, and VIII, to survey station IX, 40 in. below the floor, and thence upward to the ground surface.

Fig. 9 also gives the heat flow through the inside surface of the walls and floor as measured by the heat flow meters at locations *A*, *B*, *C*, *D*, *E*, *F* and *G*, around this perimeter. The heat flow curves are extrapolated upward to the surface of the earth, this point being plotted as the calculated heat flow through the top part of the wall if the earth in contact with it at that point had been at temperatures observed for the earth surface at survey locations III and IX, respectively.

Table 1 gives the heat flow through different sections of the basement wall on February 11, as determined from the heat flow curves in Fig. 9. The total heat loss for this date is 1,747 Btu per hour as compared with a loss of 13,734 calculated from the accepted assumption that the temperature of the earth in contact with the outside surface of the wall and floor would be 45 F. Temperature and heat flow patterns in and around the basement for February 11, April 11, and August 8, 1941, are given in Figs. 2, 10 and 11.

DISCUSSION OF TEST RESULTS

The data given in Figs. 2, 10 and 11 indicate the degree to which the earth is warmed around the basement. A continued drop in the earth temperature at the 72-in. level 21 ft away from the west basement wall from the time of the first observation on November 20, 1940, to April 8, 1941, is shown by the data plotted in Fig. 5. There is then a rise in the earth temperature at this point until about September 22, 1941. It is assumed that at this distance from the basement wall there was no warming of the earth due to the heat in the basement, although Figs. 2, 10 and 11 show some elevation of the temperature

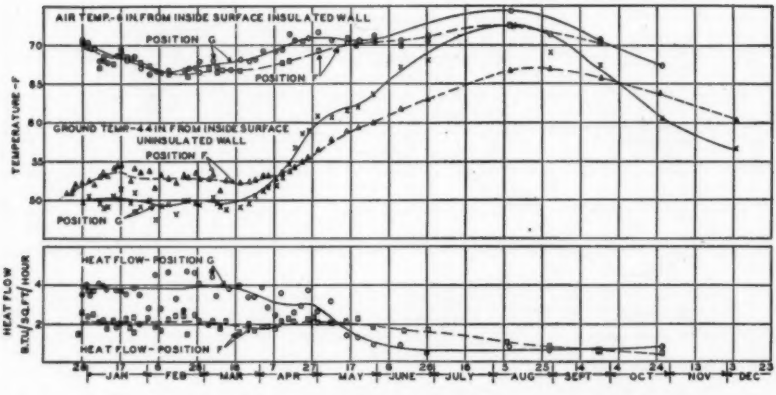

FIG. 7. RELATION BETWEEN TIME AND (1) AIR TEMPERATURE 6 IN. FROM INSIDE WALL SURFACE (2) GROUND TEMPERATURE 44 IN. FROM INSIDE WALL SURFACE, AND (3) HEAT FLOW FOR POSITIONS *F* AND *G* ON UNINSULATED WALL

in survey stations II and X at distances of 9 ft from the basement, which would indicate that some small effect extends considerably beyond this distance.

The tempefature at the 72-in. level, station I, Fig. 5, shows an annual cycle with no observable response to shorter periodic weather fluctuations. The temperature at the 42-in. level shows little fluctuation with changing weather. At the 21-in. level some temperature variation with changing weather conditions is observed, but these changes do not follow daily weather conditions. At the 9-in. level daily changes are observed. The temperature curves for the 21 and 9 in. levels are drawn as straight lines between points since the shape of these portions of the curves could not be determined. At the 3-in. level, at the ground surface and in the air 6 in. above the surface the temperature fluctuations were so violent that the effect could not be shown in the chart and they are therefore not plotted. The air temperature and the temperature of the ground surface followed more or less the hourly changes in weather, while even at 3 in. below the ground surface short period fluctuations were observed; that is, there was a rise in the temperature following a rise in air temperature or increase in sun intensity. A characteristic crossing of the

temperature curves for different depths is observed in late March and early April. Prior to this time, or while the earth temperature was falling at the 72-in. level, the ground temperature observed at survey station I generally increased with depth, while after this period the temperature decreased with depth; the only exceptions to this rule are for the points near the earth's surface where short period weather fluctuations are effective.

It is of interest to note the somewhat higher earth temperatures at the end of 1941 compared with those for the same date in 1940. The temperatures at the 72 and 42-in. levels, Fig. 5, were 55 and 50 F on December 4, 1941, compared with 51 and 46 F., respectively, on the same date, 1940. This difference is probably due to the higher weather temperatures for the three

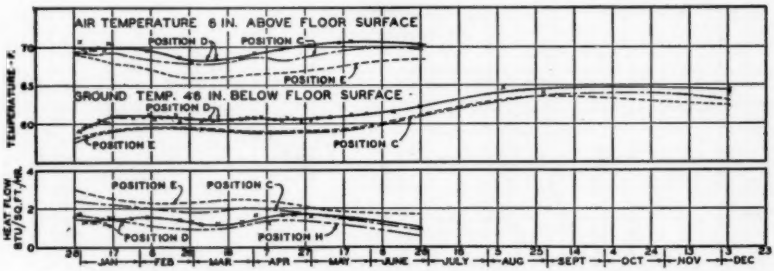

FIG. 8. RELATION BETWEEN TIME AND (1) AIR TEMPERATURE 6 IN. ABOVE FLOOR SURFACE (2) GROUND TEMPERATURE 46 IN. BELOW FLOOR SURFACE FOR POSITIONS *C, D* AND *E*, AND (3) HEAT FLOW FOR POSITIONS *C, D, E* AND *H* ON THE FLOOR

months preceding December 4, 1941, compared with the year before; from September 1 to December 4, 1941, the average daily mean temperature was 2.1 deg above normal, while for the previous year it was 2.8 deg below normal.

The relation between the temperatures at various locations around the basement for February 11, Fig. 2, indicate considerable temperature rise in the earth within 3 or 4 ft of the basement, with some appreciable effect extending to survey locations II and X, 9 ft from the basement. The effect of insulation on the west wall in maintaining a lower outside earth temperature may be observed by comparing temperatures in the survey locations III and IX, opposite the lower portions of the basement. Some of this difference, however, is due to the higher *grade line* to the east of, compared to that west of, the basement. It is also of interest to compare the inside wall surface temperatures for the insulated and uninsulated walls. The two upper inside surface temperatures of the insulated wall were 68.8 and 67.7, respectively, while for the uninsulated wall these temperatures were 65.8 and 64.4 F. respectively.

The effect of the basement in maintaining a higher temperature under the center portion of the floor is also observed. This effect seems to extend to a depth of more than 6 ft below the basement. The effect of the heated basement in heating the ground around it, and therefore decreasing the heat loss is quite pronounced, and indicates clearly the limitation of the application of the assumed 45 F for the earth temperature outside of such walls and floors.

The lag in the rising temperatures with warming of the season is indicated by comparison of Figs. 2, 10 and 11, which give data for February 11, April 11 and August 8, respectively. By April 11 the inside surface of the uninsulated wall was still around 66 F, or not more than a degree or two higher than that observed in February. This is below the dew-point temperature on an 85 F day with 55 per cent relative humidity, an atmospheric condition which may be expected on occasions at this time of the year. Hence, without dehumidification, condensation would form on this wall at times during April and on occasions throughout the early part of the summer. Even on August 8, Fig. 11, wall and floor surface temperatures in the neighborhood

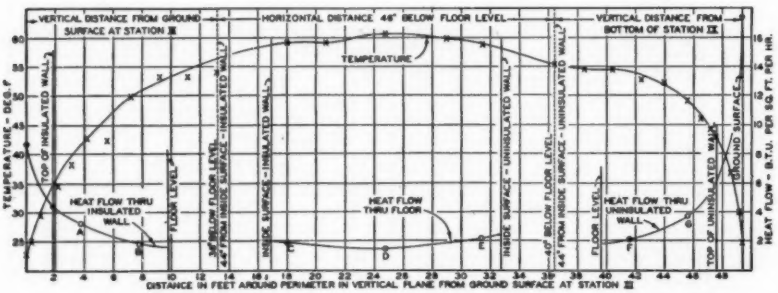
VERTICAL DISTANCE FROM GROUND SURFACE AT STATION III
HORIZONTAL DISTANCE 46" BELOW FLOOR LEVEL
VERTICAL DISTANCE FROM BOTTOM OF STATION IX
TEMPERATURE
TEMPERATURE - DEG. F
HEAT FLOW - B.T.U. PER SQ. FT. PER HR.
TOP OF INSULATED WALL
FLOOR LEVEL
36" BELOW FLOOR LEVEL
44" FROM INSIDE SURFACE - INSULATED WALL
INSIDE SURFACE - UNINSULATED WALL
40" BELOW FLOOR LEVEL
TOP OF UNINSULATED WALL
GROUND SURFACE
HEAT FLOW THRU INSULATED WALL
HEAT FLOW THRU FLOOR
HEAT FLOW THRU UNINSULATED WALL
DISTANCE IN FEET AROUND PERIMETER IN VERTICAL PLANE FROM GROUND SURFACE AT STATION III

FIG. 9. EARTH TEMPERATURES AROUND PERIMETER 44 IN. FROM INSIDE SURFACE OF SIDEWALLS AND 46 IN. BELOW FLOOR SURFACE AS OBSERVED ON FEBRUARY 11, 1941, IN SURVEY STATION III FROM GROUND SURFACE DOWN TO LEVEL 38 IN. BELOW FLOOR SURFACE IN STATIONS IV, V, VI, VII AND VIII, 46 IN. BELOW FLOOR SURFACE AND IN STATION IX FROM 40 IN. BELOW FLOOR SURFACE UP TO GROUND SURFACE

of 71 F were observed, which would result in condensation on a 95 F dry-bulb, 50 per cent relative humidity day.

Condensation did occur on the lower portion of the uninsulated wall to the extent that it was continuously damp near the corners and floor from sometime in April until close to the first of August. Noticeably less dampness was observed on the insulated wall, excepting along the floor line, where some condensation was observed. At the floor line, and more particularly on the floor near the insulated wall, there was more condensation observed than elsewhere. This may be accounted for by the lower earth temperature outside of the insulated wall due to the insulation itself and the resultant conduction of heat away from the floor around the lower edge of the insulation. From this consideration it may be assumed that more insulation, or greater resistance to heat flow, was required to avoid condensation, and that, to be effective, this added resistance to heat flow should be applied to the floor as well as to the side walls.

A comparison of the curves in Figs. 6, 7 and 8 indicates the effect of the insulation in spreading the temperature difference through the insulated wall as compared with the uninsulated wall. The effect was to increase the inside surface temperature and decrease the outside surface temperature. While the

TABLE 1—HEAT LOSS THROUGH VARIOUS PORTIONS OF THE BASEMENT WALLS AND FLOOR AS DETERMINED FROM THE HEAT FLOW CURVES, FIG. 9, FOR FEBRUARY 11, 1941

Part of Construction	Section	Insulated Portion of Walls, and Floor Adjoining Insulated Walls			Uninsulated Portion of Walls, and Floor Adjoining Uninsulated Walls		
		Area Sq Ft	Heat Loss Btu/Sq Ft/Hr	Total Heat Loss Btu/Hr	Area Sq Ft	Heat Loss Btu/Sq Ft/Hr	Total Heat Loss Btu/Hr
Side Walls	Top strip[a]	30	4.0	120	30	4.8	144
	2nd strip	30	3.5	105	30	4.0	120
	3rd strip	30	3.0	90	30	3.4	102
	4th strip	30	2.6	78	30	2.8	84
	5th strip	30	2.3	69	30	2.5	75
	6th strip	30	2.0	60	30	2.2	66
	7th strip	30	1.8	54	30	2.0	60
	Bottom strip	30	1.7	51	30	1.9	57
Floor	Outside strip[b]	19.3 3.2[c]	1.90 1.20[d]	40.5	22.3 0.3[c]	2.22 1.30[d]	49.9
	2nd strip	14.0 2.8[c]	1.90 1.20[d]	30.0	16.1 0.7[c]	2.19 1.30[d]	36.2
	3rd strip	13.6 2.3[c]	1.87 1.17[d]	28.1	14.6 1.3[c]	2.14 1.26[d]	32.8
	4th strip	13.1 1.8[c]	1.84 1.14[d]	26.2	13.1 1.8[c]	2.05 1.22[d]	29.1
	5th strip	10.6 1.3[c]	1.79 1.09[d]	20.4	9.6 1.3[c]	1.97 1.17[d]	21.6
	6th strip	8.6 0.7[c]	1.72 1.00[d]	15.5	6.5 2.8[c]	1.89 1.09[d]	15.4
	7th strip	8.3 0.3[c]	1.67 0.97[d]	14.2	5.4 3.2[c]	1.82 1.04[d]	13.1
	8th strip	6.0	1.61	9.7	6.0	1.72	10.3
	9th strip	4.1	1.58	6.5	4.1	1.68	6.9
	10th strip	2.0	1.55	2.1	2.0	1.57	2.1
	Center 0.3 Sq Ft	0.3	1.50	0.5	0.3	1.50	0.5

[a] Walls were divided into foot wide strips, extending around baseme nt parallel to floor.

[b] Floor was divided by diagonal drawn from Northeast to Southwest Corners separating portion adjoining insulated walls from that adjoining uninsulated walls; these sections were divided into foot wide strips running parallel to walls.

[c] Area of that portion of strip lying in Northeast Corner over cinders.

[d] Heat flow through that portion of strip lying in Northeast Corner over cinders.

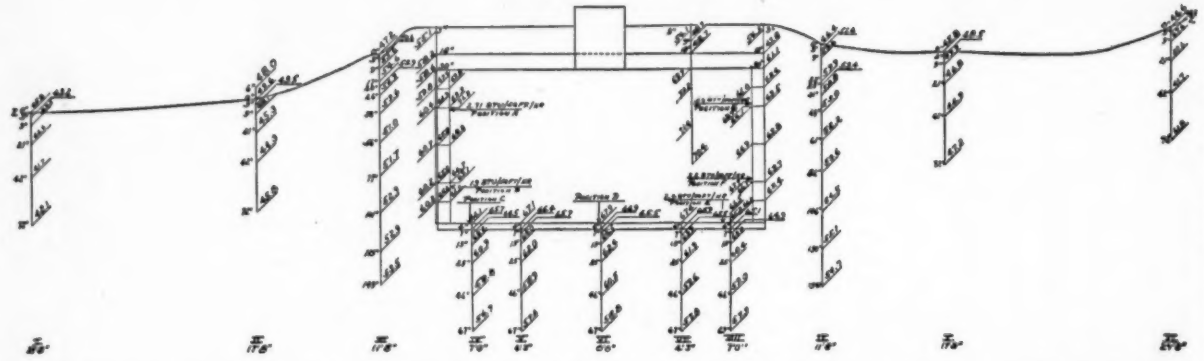

FIG. 10. TEMPERATURE AND HEAT FLOW PATTERN, SECTION ZZ, FIG. 1 FOR APRIL 11, 1941, SHOWING TEMPERATURE SURVEY STATIONS I TO XI, DISTANCES BELOW EARTH OR BASEMENT FLOOR SURFACE AND OBSERVED TEMPERATURES, ALSO NICHOLLS' HEAT FLOW METER POSITIONS *A* TO *G* AND HEAT FLOW OBSERVATIONS

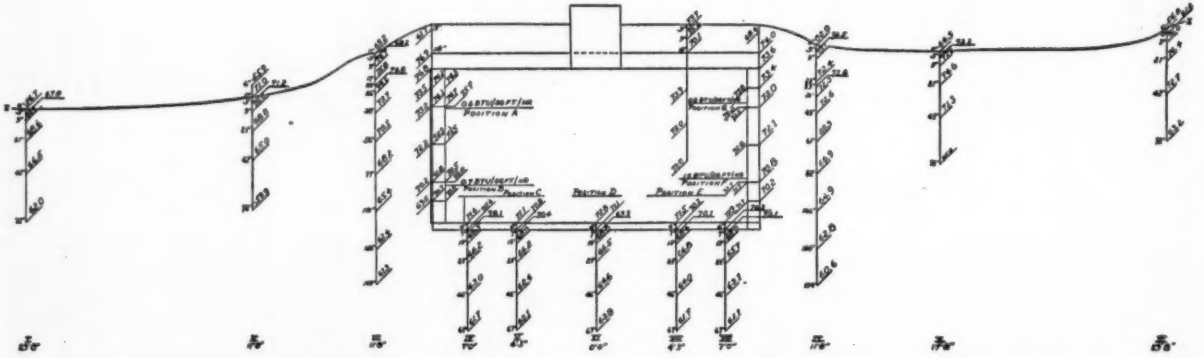

FIG. 11. TEMPERATURE AND HEAT FLOW PATTERN, SECTION ZZ, FIG. 1 FOR AUGUST 8, 1941, SHOWING TEMPERATURE SURVEY STATIONS I TO XI, DISTANCES BELOW EARTH OR BASEMENT FLOOR SURFACE AND OBSERVED TEMPERATURES, ALSO NICHOLLS' HEAT FLOW METER POSITIONS *A* TO *G* AND HEAT FLOW OBSERVATIONS

reduction in heat flow indicated through the insulated wall was not great, it was nevertheless important in reducing the noticeable condensation.

The large discrepancy between the observed heat loss and that calculated by current accepted methods is of interest, but must be taken with some reservation when applied to intermittently heated basements.

The conductivity of earth is of interest and some observations of its magnitude were made. It is obvious, however, that since there was a continual change in earth temperature, and therefore in heat storage, conductivity values could not be determined in this study with a high degree of precision. However, it was possible to make some calculations, the results of which are contained in Table 2. It will be noticed that these determinations for locations *A*, *B*, *D* and *F*, where the flow of heat should be most nearly parallel and constant, give values ranging from 6 to 8 Btu per hour per square foot per

TABLE 2—THERMAL CONDUCTIVITY OF EARTH AS DETERMINED BY HEAT FLOW THROUGH THE EARTH BACK OF OR BELOW THE METERS

POSITION	HEAT FLOW BTU/SQ FT/HR	TEMPERATURE OF OUTSIDE WALL SURFACE DEG F	EARTH		TEMPERATURE DIFFERENCE — OUTSIDE WALL TO EARTH DEG F	THICKNESS OF EARTH INCHES	CONDUCTIVITY[b]
			TEMPERATURE DEG F	LEVEL[a]			
A	3.00	58.1	41.9	38	16.2	36	6.67
B	1.70	59.6	50.0	77	9.6	36	6.37
D	1.10	66.9	60.5	46	6.4	42	7.22
F	2.00	63.1	52.9	82	10.2	36	7.05

[a] Inches below ground surface.
[b] Btu per square foot per hour per degree temperature difference for 1 in. thickness.

degree Fahrenheit temperature difference for 1 in. of thickness. It is of interest to note that this value differs little from a value of 8.5 previously reported[2] by the ASHVE Laboratory for about the same type and condition of earth. It is of further interest that the conductivity value is considerably lower for earth than for concrete. Hence, a concrete wall of a basement, backed by earth of the character studied, serves only to make an enclosure and to support the building. The surrounding earth itself offers more resistance to heat flow than the same thickness of concrete.

While this paper gives some important information on ground temperatures around a heated basement, and the effect of this warming of the ground in reducing the heat loss, it should be emphasized that these effects must be in some measure a function of the earth characteristics, including the character of the soil, the ground water level, and the possible presence of spring water flowing through the earth near the structure. In this study the earth was a red clay to a depth of 2 ft below the basement floor, and from there downward, sandstone. The basement was dug about 200 ft from a ravine cutting at least 50 ft below the level of the basement, which probably accounts for

[2] Additional Coefficients of Heat Transfer as Measured Under Natural Weather Conditions, by F. C. Houghten, Carl Gutberlet and C. G. F. Zobel. (ASHVE TRANSACTIONS, 1929, Vol. 35, p. 151).

FIG. 11. TEMPERATURE AND HEAT FLOW PATTERN, SECTION ZZ, FIG. 1 FOR AUGUST 6, 1931, SHOWING TEMPERATURE CONTOURS AT POSITIONS I TO XI, DISTANCES BELOW EARTH OR BASEMENT FLOOR SURFACE AND OBSERVED TEMPERATURES, ALSO NICHOLLS' HEAT FLOW METER POSITIONS *A* TO *G* AND HEAT FLOW OBSERVATIONS

the fact that no ground or flowing water was observed at any time at the opening especially left in the basement floor for that purpose. Undoubtedly, the temperatures and heat flows observed would have been altered by different earth characteristics, and to a great extent by the presence of a higher ground water level or seepage of cold spring water through the earth near the basement. However, while these effects would necessarily alter the magnitude of the results, the tendency observed would be the same. Whether additional studies in other locations should be made will depend upon the commercial value of the results.

DISCUSSION

P. D. CLOSE, Chicago, Ill. (WRITTEN): As has been apparent for some time, and is evident from this paper, the allowances to be made for basement heat losses should be reduced considerably from present values. While there may not as yet be sufficient data to make detailed recommendations, it seems that certain approximations are justified from the results of the paper, especially for the locality and conditions of the tests. The actual floor heat loss was less than 10 per cent of the calculated value and for the conditions of the test, the *total* heat loss averaged less than 2.0 Btu per square foot per hour through the uninsulated floor. I believe this would be a reasonable allowance to make for an average uninsulated floor for a heated basement used for living quarters or recreational purposes. For a temperature difference of say 20 deg between the air and the ground, the over-all floor coefficient even without insulation would be only 0.10. Apparently it would not be necessary to make any floor allowance for an unheated basement.

The heat loss through basement walls and floors apparently follows an entirely different pattern than is the case with walls exposed to the outside weather where steady state heat transfer is involved. With basement floors and walls there is a process of diffusion of heat into the earth and the extent of the diffusion increases as the heating season progresses until a maximum depth is reached. All of the dirt under the floor extending to the maximum limit of the heat diffusion, may be considered as contributing to the heat resistance of the floor. It appears that the depth of the diffusion of heat from the basement will increase as the conductivity of the earth increases and this increased depth of diffusion would, in turn, have the effect of offsetting the increase in conductivity of the earth.

Since the rate of heat flow through the wall varies with the depth below the ground level, the wall heat loss allowance should, of course, vary with the depth, although an average figure would probably be sufficiently accurate in most cases. The wall heat loss varied from 1.9 to 4.8 Btu per hour per square foot on the uninsulated side and from 1.7 to 3.9 per hour per square foot on the insulated side, but it should be remembered that the top of the simulated basement was covered with 18 in. of dirt so that the wall depth was greater than it would normally be. The effect of the insulation in this case is less pronounced than it would be in the case of an exposed wall because of the insulating value of the considerable thickness or depth of dirt. It would seem, however, from these tests that the *average* allowance for the wall heat loss (below grade) for a basement heated to 70 F need not be more than about 3.0 Btu per hour per square foot or a *maximum* of about 5.0 Btu per hour per square foot at the top of the grade level.

E. C. LLOYD, Lancaster, Pa. (WRITTEN): The results confirm the practice that no heat loss need be figured for floors over unheated basement spaces. Also they warrant a reduction in the heat loss figured through floors of basements used for home activities.

Admitting that these investigations have covered only a small simulated basement and that somewhat different results might come with varying soil conditions and changing moisture conditions in the soil due to seasonable changes, drainage and water levels more or less constant in location, it would seem that the results secured thus far could be applied to varying soil and moisture conditions if conductivities were available for representative soil samples in wet and dry condition. It would seem that the procurement of necessary soil samples and the determination of a K value for them dry and with varying water content, or possibly simply dry and saturated with water, should not represent an undue amount of work to supplement the excellent start already made.

The increase in conductivity of soils by reason of moisture has been brought out by earlier experimentation, but it is unfortunate that a better description of the materials previously tested is not available. A summary of soil conductivities other than those mentioned in the paper may be given as follows:

Soil, dry — $K = 0.958$	Lees & Chorlton—*ASRE*-Ins. Comm. Rep. 1922
Soil, wet — $K = 4.64$	Lees & Chorlton—*ASRE*-Ins. Comm. Rep. 1922
Soil, normal condition (excavated in Munich) including stones 1 to 3 in.	Density 127.4 lb/cubic feet K (68 F) 3.63 Groeber
River Sand, fine grained, dried completely by heat	Density 94.8 lb/cubic feet K (68 F) 2.26 Groeber
Same, with normal moisture content (6.9 per cent by weight, 11 per cent by volume)	Density 102.4 lb/cubic feet K (68 F) 7.825 Groeber

That the range in results is considerable is evident, but it may develop on further test that the differences except for those brought about by varying moisture content are inconsequent.

The importance of further consideration of condensation on the inside of these walls is a point that may well be emphasized. The increasing use of basement space for game rooms, rumpus rooms, home work shops, etc., as well as the trend toward air conditioning of such spaces brings condensation troubles in basements to the front. Some study of the effect of the addition of insulating finishes to the interior wall might well be considered at this time.

C. H. PESTERFIELD, East Lansing, Mich. (WRITTEN): I feel that we are indebted to the Research Laboratory and the authors for the work that has been done on this subject, but I feel, as probably Dr. Houghten does, that this paper is not the final solution to this problem. It is doubtful if any precise solution can ever be made due to the magnitude of variables existing in work of this nature.

It is unfortunate that the test room was not made somewhat greater in wall dimensions, for my experience on this subject has shown, for a heated basement, that at a distance of 8 ft from the outside walls that little, if any, heat is actually lost through the floor. Do not curves for position D in Fig. 8 in this work pretty fairly indicate the same conclusion?

Is it not a fact, Dr. Houghten, that it would be impossible to place the thermocouples in an isothermal plane due to the fact that the actual isothermal is inclined and not horizontal? Further, that the greater the distance from the wall in either direction, the plane changes angle.

It is gratifying to note that the data in this paper substantiate the belief that many of our engineers have had for years, that the heat loss through the basement has been grossly exaggerated; however, few expected that the actual loss and the calculated loss differed so greatly.

E. F. DAWSON, Norman, Okla. (WRITTEN): This paper presents information that is a very important addition to our present building heat loss data and places on a scientific basis the determining of design heat losses through basement walls and floors. These heat loss calculations up to the present have, in many cases, been made on assumptions based chiefly on guess work or incomplete test data.

The test conditions were, in general, quite representative. The climatic conditions at the test location, Pittsburgh, Pa., are representative of many sections of the country.

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The test room built for this research project, with its side walls entirely below grade, is typical of many basement rooms and below grade rooms of large buildings. The use of both insulated and uninsulated walls and floors in the construction of this test room furnishes valuable data for comparing heat flow, temperature and condensation effects.

The graphical results shown in Fig. 5 present interesting and valuable information relative to the soil temperatures at depths from 9 to 72 in. for a 12 month period. These observations made at a station 21 ft from the structure wall were not noticeably influenced by the heated basement room, and hence might serve as a guide in estimating soil temperatures for intermittent heating design. Also, by comparing similar data at other locations with these soil temperatures we will be aided in our heat loss estimates at these locations.

The information portrayed on Figs. 6, 7 and 8 giving the logs of the air temperature 6 in. from the inside surfaces and the soil temperatures 44 in. from these surfaces will be invaluable in studying air to soil temperatures and arriving at representative temperature differentials for heat loss determinations through basement walls and floors.

The soil temperature gradient around the perimeter of the structure together with the heat flow at 7 observation points is shown in Fig. 9 for a typical heating design day. The soil temperature values in the vertical stations of this perimeter may be useful in estimating soil temperatures of other basement rooms whose walls extend to a greater or less depth below grade level.

The discrepancy between the *rule of thumb* method, which gave 13,730 Btu per hour as the heat loss for this room and the determined heat loss by test of 1750 Btu per hour is such as to indicate the great need for this more scientific method of approach.

The variables that will need to be considered for the general application of basement room losses are: soil characteristics, soil temperature gradients in warmer and colder localities and basement rooms that have part of their walls above grade level.

The elimination of condensation appears to be, as stated by the authors, a matter of applying additional heat resisting materials to the walls and floors.

A. B. ALGREN, Minneapolis, Minn. (WRITTEN): I would like to commend the authors on the contribution they have made in starting to clarify the question as to ground temperatures. Although the results obtained would possibly apply to many localities, they also indicate that further studies are necessary and should, if possible, be made with respect to different geographical locations. A ground temperature of 50 F, a value that we assume for heat loss calculations, appears to be about correct for our locality. This again is based more upon assumption than actual test values.

I do feel that this work should be extended to other localities, not only for its value in accurately calculating basement losses, but mainly with respect to the condensation problem in basements. With definite data on ground or earth temperatures, the mechanics required for eliminating condensation would then become quite simple.

DR. HOUGHTEN: Answering Professor Pesterfield: It is true that the isothermal planes surrounding the basement are not horizontal. About 2 ft of the thermocouples in the bottom of the holes were coiled into horizontal spirals and therefore not strictly in the isothermal plane, although not far from it.

No. 1214

SEMI-ANNUAL MEETING, 1942

St. Paul, Minn.

THE Semi-Annual Meeting 1942 of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS was the main attraction for 200 members and guests, who came to St. Paul, June 15-17, for a discussion of papers, which were not only of current interest, but provided fundamental information that is useful to engineers engaged in war work, involving heating, ventilating and air conditioning.

At 2:00 P.M. on Monday, June 15, the first technical session was called to order by Pres. E. O. Eastwood, and William McNamara, president of the Minnesota Chapter, gave a word of greeting, and then introduced Frank Madden, secretary to Mayor McDonough, who made a speech of welcome on behalf of the City of St. Paul. He outlined, in a most entertaining way, some of the many attractions that awaited the members.

A brief response was made by President Eastwood, who expressed the appreciation of the Society for the cordial greeting from the City officials and the people of St. Paul.

President Eastwood then called for the presentation of the technical papers dealing with performance of a forced warm-air heating system as affected by changes in volume and temperature of air recirculated, effect of vanes in reducing pressure loss in elbows in 7-inch square ventilating duct, and trend curves for estimating performance of panel heating systems.

W. L. Fleisher expressed the thanks and appreciation of the membership for the valuable contribution which resulted from these cooperative research studies.

Before adjournment R. E. Backstrom, General Chairman of the Committtee on Arrangements for the meeting, announced the program of entertainment events and invited all members to participate.

The second technical session was called to order by President Eastwood at 9:30 A.M., Tuesday, June 16, and he introduced First Vice-Pres. M. F. Blankin, who conducted the meeting.

He introduced the authors of the technical papers, which covered overloading of viscous air filters during accelerated tests, and air conditioning of blackout plants.

Mr. Fleisher made a suggestion that a special committee should be formed to study the air conditioning question of blackout plants.

Second Vice-Pres. S. H. Downs introduced Lt.-Comdr. A. R. Behnke (MC) U.S.N., who presented an interesting address on physiologic studies pertaining to Naval ventilation. At the conclusion of a very interesting discussion of

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this subject, Mr. Downs introduced Dr. Charles Sheard, director of the Division of Biophysical Research of Mayo Foundation. Dr. Sheard's subject was on some further investigations of calorimetric and skin temperature measurements of the extremities and their significance.

The third and final technical session was held at 9:30 A.M. on Wednesday, June 17, and was opened by President Eastwood. He announced that the technical papers would follow on the subjects of heat loss studies in four identical buildings to determine the effect of insulation, the specific heat of thermal insulating materials, and on the simultaneous heat and vapor transfer characteristics of an insulating material.

At the conclusion of the technical program W. H. Driscoll, past president of the Society, paid tribute to the past presidents who had died within a few months. He said that it is a strange and sad fact that the Society has lost three of its very prominent past presidents and its last honorary member within a short time. Not in the history of the Society had such a tragedy occurred, and he felt that it was appropriate to recall some of the contributions that had been made by these gentlemen, so that the younger members might fully appreciate the extent of their activities, not only in their own, but in other fields of endeavor.

REGINALD P. BOLTON (October 5, 1856-February 18, 1942) *Honorary Member and Life Member* was, perhaps, one of the most versatile men who ever served the Society. He was a most remarkable man and his interests extended into many fields and his engineering activities took him to many foreign countries. He was a great historian and recorded the history of Manhattan Island, and was recognized as an expert on Indian lore. Mr. Bolton was a dynamic and fascinating individual whose long and active career was an inspiration to all who knew him.

JOHN HALE (November 11, 1870-May 29, 1942), *Life Member* was actively interested in the Society's work throughout his life. He was of an aggressive, bright and friendly personality. He was regular in his attendance at Society meetings and had the unusual distinction of serving as the Society's chief executive for two years, one year as acting president while the late John R. Allen was in Turkey, followed by his regular term as president of the Society. He was a fine committee member and served the Society well. The members will miss John Hale.

J. IRVINE LYLE (February 14, 1874-June 7, 1942), *Life Member* was one of the great characters in the Society and one of the small group who had the vision and the energy to conceive and bring into operation one of the Society's greatest endeavors, the Research Laboratory. He contributed many technical papers to Society meetings, and maintained an active interest in its Chapter work. He had wide interests in many other fields but always maintained a close contact with the work of the Society. He was a big man, striking in appearance, dignified, human and understanding. He was a great credit to the engineering profession and a staunch friend.

It is a great privilege to pay this tribute to all of these men who served the Society so well.

President Eastwood called on the Chairman of the Resolutions Committee, T. D. Stafford, who presented the following report:

Report of Committee on Resolutions

WHEREAS, the Semi-Annual Meeting 1942 of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has been an outstanding success in the City of St. Paul, Minnesota, June 15-17, 1942.

THEREFORE, BE IT RESOLVED, THAT an expression of thanks and appreciation be adopted by this meeting and be spread upon the minutes of the Society and copies

thereof be transmitted to each of the persons and agencies who have contributed to make this meeting so enjoyable for the members of the Society who attended:

To the Minnesota Chapter members for the capable manner in which they fulfilled their position as hosts;

To Chairman R. E. Backstrom and the Committee on Arrangements whose careful and detailed planning have been revealed in every department of the meeting;

To Mrs. Backstrom and her Committee for their splendid entertainment of our ladies and to the citizens who kindly cooperated in welcoming them to inspect their gardens and those who furnished their cars for the trips;

To the Authors and Speakers at Technical Sessions for their instructive papers and able presentations;

To Dr. Walter H. Judd for his thorough analysis of our national crisis;

To newspapers and trade publications, whose columns have given advance notices and daily coverage of this Semi-Annual Meeting;

To the M. & O. Paper Co. for the able assistance of Leonard Costley (Paul Bunyan);

To Hal Olson and his talented assistants, Messrs. Algren and Jones, for their impersonations and feats of magic;

To E. H. Pitney of the War Production Board for his interesting address;

To Victor Johnson, Secretary of the Minnesota Tourist Bureau for the entertaining movies;

To the Wood Conversion Co. for the inspection and entertainment at the First National Bank Tower;

To the Faculty of the Engineering Department of the University of Minnesota, for their aid in arranging and conducting the tour of inspection;

To the management and all employees of the Hotel St. Paul for their unusually courteous and cooperative efforts;

To the St. Paul Association of Commerce for the assistance rendered by their Convention Bureau representatives;

To Mayor John J. McDonough and his Executive Secretary, Frank Madden, and to the various Municipal Departments and citizens of St. Paul for their splendid aid and cooperation.

AND WHEREAS the Society's world-wide influence; its advancement of technologic knowledge in the science of heating, ventilating, and air conditioning, to its present high level; the organization of its members into one friendly, firmly cemented, altruistic, cooperative union, culminating in successful meetings of lasting progress and innumerable endearing memories, has been accomplished only through the interest, unstinting labors, planning, and unselfish devotion to principle and mutual assistance of the Founders and former Officers of the Society;

THEREFORE, BE IT FURTHER RESOLVED, THAT an expression of congratulation and felicitation be adopted by this assembly and spread upon the minutes of the Society, and that copies thereof be transmitted to each of the Charter Members and Past Presidents of the Society:

To congratulate all Charter Members and Past Presidents on the salutary results of their efforts;

To express our pleasure at the attendance of so many Life Members and Past Presidents; and

To express to Charter Members and Past Presidents who could not be in attendance our sincere regret of their inability to be with us on this occasion.

Respectfully submitted,

THOMAS D. STAFFORD, *Chairman*
C. S. KOEHLER
J. DONALD KROEKER

Mr. Backstrom responded to the vote of thanks and appreciation which had been unanimously adopted, and then told of the pleasure of the members of the Minnesota Chapter in having the Society visit the Twin Cities.

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President Eastwood announced that the registration had reached a total of 200, after which the meeting adjourned.

Council Meeting

A meeting of the Council, with 16 of the 18 members present, was held on Monday morning, June 15, at the Hotel St. Paul, and President Eastwood presided. Reports of Council committees indicated that the present membership status on June 1 was 3167. The Finance Committee presented some amendments to the budget on the basis of operations during the first five months of the year. The budget, exclusive of research, of $87,600.00 was approved.

A detailed discussion of research activities indicated that the program for the future would be devoted only to problems that might contribute to winning the war.

Nomination of five members to serve for three-year term on Committee on Research, beginning January 1943, was made as follows: H. J. Rose, Pittsburgh; L. P. Saunders, Lockport, N. Y.; L. E. Seeley, New Haven; A. E. Stacey, Jr., Washington, D. C.; and C. Tasker, Toronto. The resignations of nine members were accepted reluctantly and the membership of 132 members was cancelled for non-payment of dues.

Meeting of Chapter Representatives

On June 15, at the request of the Chapter Delegates present, Chairman J. F. Collins, Jr., called a meeting and the following were present: Albert Buenger, Cincinnati; C. E. Bentley, Golden Gate; E. M. Mittendorff, Illinois; A. L. Walters, Iowa; E. K. Campbell and W. A. Russell, Kansas City; M. B. Shea, Michigan; W. R. White, Nebraska; C. S. Koehler, New York; J. D. Kroeker, Oregon; H. B. Hedges, Philadelphia; T. F. Rockwell, Pittsburgh; T. D. Stafford, Western Michigan.

Mr. Kroeker reported that the threatened fuel shortage in the Pacific Northwest had caused the municipal authorities in the City of Portland to enlist the assistance of the Oregon Chapter to prepare a report and give publicity to fuel conservation measures.

The various chapter representatives outlined the methods being used in establishing dues rates for members in service.

Chairman Collins pointed out that all chapter officers should be advised that chapter rolls could include only members of the Society in good standing. The delegates present requested that the chairman communicate with all chapters in this matter.

It was announced that the Local Chapters Committee would be able to assist chapters that needed speakers this fall and letters should be mailed by the chairman to determine chapter requirements.

Entertainment

On June 14, early arrivals for the Semi-Annual Meeting of the Society were received by the Reception Committee in the Capitol Room of the St. Paul Hotel. At 6:30 on Sunday evening 75 members and guests gathered to enjoy the special smorgasbord dinner in the Club Casino, St. Paul Hotel. Follow-

ing the dinner members enjoyed a motion picture showing Kodiak Bear hunting.

At noon on Monday, June 15, a get-together luncheon was held in the Club Casino with 145 in attendance, and E. H. Pitney of the Contract Distribution Branch of the War Production Board, gave an interesting discussion of the work.

During the day many of the visitors were mystified by the appearance of a *bearded giant* who carried a logger's ax, but the mystery was solved on Monday evening in the Club Casino when *Paul Bunyan* (Leonard Costley) told his story without exaggeration. The Minnesota members appeared in lumber jackets and boots for this occasion, and the Club Casino was set in logging camp style with long tables, tin plates and mountains of food. During the meal Mr. Costley told the story of *Paul Bunyan,* "who was a mythical hero of the lumber camps of the Northwest. The stories which have been told in American timberlands for generations are of Canadian origin, but American loggers below the border made of *Paul Bunyan* a true hero of camp nights' entertainment.

"They gave him Babe, the blue ox, who measured 42 ax handles and a plug of chewing tobacco between the horns. They created the mythical logging camp with the cookhouse of mountainous size, and they peopled this camp with astounding minor heroes.

"They made their *Paul Bunyan* an inventor and an orator and an industrialist whose labors surpassed those of Hercules. They devised a chronology for him; he ruled American life in the period between the *Winter* of the *Blue Snow* and the *Spring* that the *Rain Came Up from China.*

"In any case, the legend which is said to have begun in the Papineau Rebellion in Canada in 1837, when a *mighty-muscled, bellicose, bearded giant named Paul Bunyan raged among the Queen's troops like Samson among the Philistines,* had by 1860 spread throughout the Northwest, and perhaps even into the South where under the name of John Henry he became *the man worth talking about* in the work camp gang.

"Some evidence points to a French-Canadian origin among the loggers of Quebec or northern Ontario. Other evidences point just as strongly to an American beginning, possibly in Michigan or Wisconsin. In the Minnesota camps, the legends were undoubtedly enriched by Scandinavian myth, and there *Paul Bunyan* became a sort of modern Thor."

An unexpected feature of the program came when Paul carried in a miniature wedding cake with one candle and presented it with appropriate ceremony to Mr. and Mrs. George Hatch, Toronto, to celebrate the first week of their marriage.

The next entertainment feature was introduced by E. F. Jones, chairman of the Entertainment Committee, who presented Hal Olson, an impersonator and magician extraordinary. In some of his card tricks and illusions, he had the assistance of Prof. A. B. Algren, who literally lost his shirt on one occasion.

Following the entertainment a special sound movie showing the historic lakes and scenic features of the State was presented by Victor Johnson, secretary of the Minnesota Tourist Bureau.

A fitting climax for the entertainment program was provided by the dinner and dance at the St. Paul Hotel on Tuesday evening, June 16. Hors d'oeuvres

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were served in the Club Casino and a fine dinner followed in the Continental Room. Toastmaster H. E. Gerrish presented the officers, past presidents and distinguished guests at the head table. W. T. Jones, Boston, presented a memory book to Walter L. Fleisher, who was president of the Society in 1941. Mr. Fleisher expressed his thanks in the form of verse as follows:

Memory is a fickle thing
  One day you think you are king
The next it's gone, perhaps it's Spring
  And flowers crowd or rather cloud
Those happy days you said aloud
  You'd ne'er forget.
And so it's good when work is done
  And all deeds that you did are sung
To have a book in which is brought
  The things and acts the papers caught,
So as you age and hair grows grey
  To children and grandchildren say,
Those are the things I did!
  Pandora, raise not the lid
You may not find, all thoughts are kind
  So I forbid!
But no—on books do not rely
  The written word may seal a lie!
I would not hold this book so dear
  Unless in *you*, my memory's clear.

F. C. McIntosh, Chairman of Committee on Research, presented the Research Cup to Roy E. Gorgen, Minneapolis, who had a low net of 69. The Eichberg Memorial Cup was presented by M. F. Blankin to Capt. E. F. Bell of the Minnesota Chapter team, consisting of Messrs. R. E. Gorgen, E. L. Knowles and E. F. Bell. Mr. Bell, Chairman of the Golf Committee, announced the names of others who were rewarded for their efforts on the golf course: *Low Gross*—Arthur Nutting; *Low Net Gross*—A. O. May, 71; M. F. Blankin, 77; G. B. Priester, 78; Richard Ordway, 78; J. H. Milliken, 78; F. H. Gaylord, 79; J. C. Matchett, 80; M. W. Bishop, 80; F. H. Schernbeck, 83; A. B. Algren, 85; B. L. Casey, 86; and C. E. Price, 86.

Blind Bogey: V. J. Jenkinson, 77; H. E. Gerrish, 76; and E. Anderson, 83. *High Gross:* Albert Buenger, 131. *Most 8's:* L. T. Avery, 4; and A. L. Byfield, 4. *Most 6's:* William Strum, 6. *Most 7's:* N. D. Adams, 6.

The principal speaker, Dr. Walter H. Judd, was introduced and gave a stirring talk on the subject *Behind the Conflict in the Pacific.* He recalled some of the incidents of his life in China and gave a clear insight into the methods of thought followed by the Orientals. At the conclusion of Dr. Judd's speech, the room was cleared for dancing.

As soon as the ladies registered they were greeted by the Ladies Committee of the Minnesota Chapter under the direction of Mrs. R. E. Backstrom. Tea was provided on Monday afternoon to 42 ladies at the Women's City Club, St. Paul, and they enjoyed an interesting talk.

On Tuesday morning private cars were provided for sightseeing trips in the Twin Cities and 44 ladies enjoyed luncheon at the Coffman Memorial Union at the University of Minnesota, and then toured the Minnetonka Lake district and visited several formal gardens on private estates.

On Wednesday 40 ladies enjoyed a luncheon and card party. There were eight tables and the prize winners were: Mrs. E. M. Mittendorff, Chicago;

Mesdames E. Uhl, C. T. Lawrence, E. L. Knowles, F. H. Schernbeck, A. B. Newton, A. W. Schultz, all of Minneapolis; and Miss Dorothy Matchett, Chicago; Mrs. R. C. Colman, St. Paul and Mrs. C. O. Carlson.

PROGRAM SEMI-ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
ST. PAUL HOTEL, ST. PAUL, MINN.
June 15-17, 1942

June 14

3:00 P.M. REGISTRATION AND RECEPTION OF EARLY ARRIVALS (*Capitol Room*)
4:00 P.M. War Service Committee Meeting.
6:30 P.M. Smorgasbord Dinner (*Club Casino*)

June 15

8:30 A.M. Meeting of Committee on Cooling Towers, Evaporative Condensers and Spray Ponds
9:00 A.M. REGISTRATION (*Capitol Room*)
Minnesota Chapter and Ladies' Headquarters (*Sibley Room*)
9:30 A.M. Council Meeting (*Room 234*)
10:00 A.M. Nominating Committee Meeting (*Room 334*)
10:00 A.M. Meeting of Committee on Code for Testing and Rating Air Cleaning Devices
12:15 P.M. Get-Together Luncheon (*Club Casino*)
1:30 P.M. TECHNICAL SESSION (*Continental Room*)
Official Greetings by Frank Madden
Performance of a Forced Warm-Air Heating System as Affected by Changes in the Volume and Temperature of the Air Recirculated, by A. P. Kratz and S. Konzo
Effect of Vanes in Reducing Pressure Loss in Elbows in 7-Inch Square Ventilating Duct, by M. C. Stuart, C. F. Warner and W. C. Roberts
Trend Curves for Estimating Performance of Panel Heating Systems, by B. F. Raber and F. W. Hutchinson
2:30 P.M. Ladies' Tea and Speaker—St. Paul Women's City Club
4:30 P.M. Committee on Research (*Room 234*)
7:30 P.M. Paul Bunyon Dinner (*Club Casino*)

June 16

9:30 A.M. TECHNICAL SESSION (*Continental Room*)
Overloading of Viscous Air Filters During Accelerated Tests, by F. B. Rowley and R. C. Jordan
Air Conditioning of Blackout Plants, by W. A. Grant
Address—Physiologic Studies Pertaining to Naval Ventilation, by A. R. Behnke, Lt. Comdr., (MC) U.S.N.
Address—Some Further Investigations on Calorimetric and Skin Temperature Measurements of the Extremities and Their Significance, by Dr. Charles Sheard
10:00 A.M. Ladies' Sightseeing Trip—Twin Cities
NOON Ladies' Luncheon at Coffman Memorial Union, University of Minnesota, and Ladies' Sightseeing Trip—Minnetonka Lake District
12:15 P.M. Men's Luncheon at White Bear Yacht Club
1:30 P.M. Golf Tournament (Research Cup and Eichberg Memorial Cup)
2:00 P.M. Meeting of Committee on Air Distribution and Air Friction
2:00 P.M. Meeting of Committee on Air Conditioning in Industry
6:45 P.M. Pre-Banquet Get-Together (*Club Casino*)
7:45 P.M. Dinner and Dance (*Continental Room*)
Toastmaster: Harry E. Gerrish
Speaker: Walter H. Judd, M.D. *Subject:* Behind the Conflict in the Pacific
Presentation: Past President's Memory Book to W. L. Fleisher

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June 17

9:30 A.M. TECHNICAL SESSION (*Continental Room*)
Heat Loss Studies in Four Identical Buildings to Determine the Effect of Insulation, by D. B. Anderson
The Specific Heat of Thermal Insulating Materials, by G. B. Wilkes and C. O. Wood
Simultaneous Heat and Vapor Transfer Characteristics of an Insulating Material, by F. G. Hechler, E. R. McLaughlin and E. R. Queer
12:00 P.M. Ladies' Luncheon, Card Party, and Golf
12:15 P.M. Men's Luncheon at Somerset Country Club
1:30 P.M. Golf Matches and Sightseeing and Inspection Trips

COMMITTEE ON ARRANGEMENTS

Minnesota Chapter

R. E. BACKSTROM, *General Chairman*
A. B. ALGREN, *Vice-Chairman*

Executive

N. D. ADAMS H. E. GERRISH F. B. ROWLEY

Banquet—H. M. BETTS, *Chairman*
Entertainment—E. F. JONES, *Chairman*
Finance—M. S. WUNDERLICH, *Chairman*
Transportation—J. F. STAFFORD, *Chairman*
Reception—M. H. BJERKEN, *Chairman*
Publicity—D. B. ANDERSON, *Chairman*
Ladies—MRS. R. E. BACKSTROM
Golf—E. F. BELL, *Chairman*

Board of Governors

WILLIAM MCNAMARA G. A. DAHLSTROM

No. 1215

PERFORMANCE OF A FORCED WARM-AIR HEATING SYSTEM AS AFFECTED BY CHANGES IN VOLUME AND TEMPERATURE OF AIR RECIRCULATED

By A. P. KRATZ * AND S. KONZO,** URBANA, ILL.

INTRODUCTION

CONSIDERABLE flexibility exists in the choice of air volumes and air temperatures to be used in the design of a forced warm-air heating system. For a given structure, the design of a duct system may be based on the use of either a large volume of air at a relatively low register air temperature or a smaller volume of air at a higher register air temperature. Furthermore, after the system is installed, satisfactory performance may be obtained even though the actual combinations of air volumes and register air temperatures encountered deviate widely from those assumed for the design of the duct system.

This flexibility in conditions affecting both the design and the operation of a forced warm-air heating system may be considered as an advantageous feature, inasmuch as any reasonable deviations in the operating conditions from those assumed for the design can be readily compensated for by relatively minor changes in the plant. On the other hand, this extreme flexibility in the choice of both design and operating conditions has resulted in the development of many *rule of thumb* methods and unconfirmed theories.

Proper development of any method of heating which is undergoing continual evolution and improvements in technique, requires that basic facts be substituted for *rules of thumb*. One of the prevailing theories of design is that in the case of a forced warm-air system installed in a well insulated structure, a certain minimum number of air recirculations, commonly referred to as air changes, is essential for satisfactory operation. For the purpose of determining whether or not the plant performance as affected by independently varying the quantity and temperature of the air circulated had an adverse effect on the environment produced, several series of tests were conducted

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Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, St. Paul, Minn., June, 1942.

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in the Warm-Air Research Residence during the winter seasons of 1939-1940 and 1940-1941. In the first season the tests were made with the 12 warm-air registers located in the baseboard, and in the second season the tests were repeated with the same number of warm-air registers located in the sidewalls and approximately 7 ft above the floor.

In order to make direct comparisons between the results obtained with the various methods of operation, only one change at a time was made in the items specifically under observation; namely, register location, air volume, or the setting of the bonnet thermostat.

DESCRIPTION OF RESEARCH RESIDENCE AND HEATING EQUIPMENT

The Research Residence and the forced-air heating plant have been completely described in a previous publication.[1] The Residence is a three-story structure of standard frame construction with side walls and ceiling fully insulated[2] with mineral wool insulation. The total space heated, including a sun room but not including the basement, amounted to 17,540 cu ft. All of the 50 windows on the three stories of the Residence, with the exception of two small quarter-round windows in the east dormitory, were provided with tightly fitting storm sash. The calculated heat losses were approximately 51,100 Btu per hour at an indoor-outdoor temperature difference of 80 F. The Residence is completely furnished, and during the heating season it was occupied by five people.

The heating plant consisted of a cast-iron, circular-radiator warm-air furnace used in connection with a forced-air heating system. The furnace was fired by means of a coal stoker of the underfeed type. The fuel burned was 1-in. x 10-mesh stoker coal, washed and oil treated, and was obtained from Saline County, Ill.

The heating plant was controlled by means of a room thermostat operating to start and stop the stoker motor, and to start and stop the circulating fan. This room thermostat was of the heat-anticipating type, and was used in conjunction with two bonnet thermostats which served as high and low limit controls for the temperature of the air in the furnace bonnet. When the room thermostat operated to start the fan and stoker, one bonnet thermostat (fan control) prevented the fan from starting unless the bonnet temperature was above 125 F, except for the series of tests in which the bonnet thermostat settings were changed in order to determine the effect of varying the air temperature. Similarly for all tests, except those in which the bonnet thermostat settings were changed, the other bonnet thermostat (limit control) prevented the stoker from starting unless the bonnet temperature was below 150 F. Both the fan control and limit control were incorporated in one container and the adjustment of the thermostat indicator changed the setting of both controls simultaneously. The temperature differential between the fan cut-in point and the fan cut-out point was approximately 40 F.

During the season of 1939-1940 the 12 warm-air registers in the Residence were located in the baseboard and were of the sizes shown in Table 1. The

[1] University of Illinois, *Engineering Experiment Station Bulletin No. 266*, by A. P. Kratz and S. Konzo, 1934.

[2] Effect of Insulation on Plant Performance in the Research Residence, by A. P. Kratz and S. Konzo. (ASHVE TRANSACTIONS, Vol. 47, 1941.)

registers were equipped with directional vanes and directed the air sidewise and parallel to the floor. Due to the fact that the register locations were dictated by the location of warm-air stacks which had originally been installed in the Residence in conformity with certain structural features, the arrangement cannot necessarily be considered as the best for each room. An improvement in the air distribution might have resulted had it been possible to use the most favorable location of the register in each case, but tests to verify such possibilities were not considered as within the scope of the present investigation. During the season of 1940-1941 a stack extension was added to an opening in the top of each of the baseboard stackheads and a new stackhead was located at the top of the extension and approximately 7 ft above the floor.

TABLE 1—SIZES OF WARM AIR REGISTERS

ROOM	BASEBOARD[a] REGISTERS, INCHES	HIGH SIDEWALL REGISTERS, INCHES	STACK SIZE ABOVE BASEBOARD REGISTERS, INCHES
Living	12 × 6	14 × 5	3¼ × 14
Hall	14 × 4	14 × 4[b]	3¼ × 14
Dining	12 × 6	12 × 4	3¼ × 12
S. Sunroom	12 × 10	12 × 4	3¼ × 12
N. Sunroom	12 × 10	12 × 4	3¼ × 12
Kitchen	14 × 10	14 × 4	3¼ × 14
E. Bedroom	12 × 6	12 × 6	3¼ × 12
S. W. Bedroom	12 × 6	12 × 4	3¼ × 12
Bath	10 × 8	10 × 4	3¼ × 10
N. W. Bedroom	12 × 10	12 × 4	3¼ × 12
E. Dormitory	12 × 10	10 × 4[b]	3 × 10
W. Dormitory	12 × 10	10 × 4[b]	3 × 10

[a] Description given in University of Illinois, Engineering Experiment Station Bulletin No. 318, pages 15 and 16.
[b] Registers approximately 6 ft above floor, all others 7 ft above floor.

Sidewall registers, of the sizes shown in Table 1, were installed with the register faces flush with the plaster walls. The registers directed the air parallel to the floor with no sidewise deflection. The nearest commercial sizes of sidewall registers were selected, consistent with a register air velocity of approximately 600 fpm based on a total air recirculation of 1675 cfm. The sidewall registers were considerably smaller than the corresponding baseboard registers for which the register air velocity averaged approximately 300 fpm. During the entire series of tests no changes were made in the return air system, which included grilles in the living room, hall, and sun room.

METHOD OF CONDUCTING TESTS

The average of the air temperatures in all of the rooms of the house was maintained at 72 F at the 60-in. level both day and night. Observations of weather, indoor room air temperatures, room relative humidities, and other incidental data were made daily at 7:00 A.M., 11:00 A.M., 4:00 P.M., and 10:00 P.M. Complete data were obtained for each 24-hour test period on the fuel

consumption, weight of ash and clinkers removed, the total integrated time of operation of the fan and of the stoker, the total electrical input to the fan and stoker motors, and the total number of *on-periods* of both the circulating fan and the stoker. Daily observations were made of the volume of air circulated, and the filters were cleaned with sufficient frequency to maintain the air volume constant. In addition, continuous records of temperatures, CO_2, and the index of smoke density were obtained for each 24-hour period. Each day at 11 A.M., the clinkers were removed, the fuel bed was levelled, and the hopper was filled with coal. During extremely mild weather no attention was given to the fuel bed or the coal in the hopper, except as required every two or three days. By means of the balanced check damper, the draft in the smoke pipe was maintained at approximately 0.05 in. of

TABLE 2—SUMMARY OF TEST SERIES

SERIES NO.	SEASON	REGISTER LOCATION	AIR VOLUME, CFM	FAN CUT-OUT POINT, F	
2–39	1939–40	Baseboard	1675	125	
4–39	1939–40	Baseboard	1300	125	
5–39	1939–40	Baseboard	800	125	150[b]
1–40	1940–41	High wall	1675	125	
7–40	1940–41	High wall	1300	125	
5–40	1940–41	High wall	800	125	154[b]
4–40	1940–41	High wall	800		137[b]
6–40	1940–41	High wall	800		179[b]
3–40[a]	1940–41	High wall	800		137[b]

[a] Full register opening resulting in average register air velocity of approximately 290 fpm, whereas all other tests with high wall registers were made with 600 fpm register air velocity.

[b] Mean bonnet air temperature determined by average of maximum and minimum values. For the tests with 800 cfm the mean bonnet temperature is of greater significance than the fan cut-out point as determined from the setting of the bonnet thermostat.

water. For each series of tests, data were obtained over a wide range of outdoor weather conditions.

For the series of tests in which baseboard registers were used, the full-sized register openings were maintained with the three quantities of air circulated, and the register air velocities decreased as the quantity of air circulated was decreased, varying from a maximum of 300 fpm to a minimum of 140 fpm. On the other hand, the series of tests in which high sidewall registers were used, with the exception of series 3-40, a register velocity of 600 fpm was maintained regardless of the total quantity of air circulated. This was done by first adjusting the dampers in the duct system to obtain the volume of air required to heat each room, and then adjusting the size of the register opening by partially covering it with a plate so that a register air velocity of 600 fpm was obtained. The series of tests run are shown in Table 2.

The air delivery of the fan, listed in Table 2, was measured by means of a Pitot tube, placed in the return duct and calibrated in position. For the purpose of calibration, prior to the start of the heating season the entrances

to the outlet ducts were blocked and all of the air delivered by the fan was passed through the furnace and out of a 15-in. diameter calibrating duct connected to the outlet side of the furnace casing. Twelve-point velocity traverses were made at a measuring station located in the calibrating duct, and the air volumes, as determined by the traverse, were correlated with the readings of the fixed Pitot tube located in the return duct. When the calibration was completed, the calibrating duct was removed and the entrances to the trunk ducts were opened in order to make the furnace ready for service. By adjusting the fan speed and the position of the dampers on the intake side of the fan, air volumes of 1675 cfm, 1300 cfm, and 800 cfm were readily obtained.

RESULTS OF TESTS

Variations in Air Volume Delivered from Baseboard Registers (*Series* 2, 4, 5-39): For each series of tests, data were obtained over a wide range of outdoor weather conditions, and were plotted as points on graphs in which the abscissae were indoor-outdoor temperature differences. A curve was drawn through the points representing each series of tests, and the values obtained from the curves at indoor-outdoor temperature differences of 34 F and 55 F were selected as typical of average and relatively cold winter days respectively. The values so obtained have been plotted in Fig. 1.

In Fig. 1 the circled points and the full line curves show the significant results obtained for baseboard registers with various air deliveries on a day in which the indoor-outdoor temperature difference was 55 F. The top set of curves, showing the maximum and minimum values of the bonnet air temperatures, indicate that, although no change was made in the setting of the bonnet thermostat, the range of bonnet air temperatures was slightly higher with low air deliveries. This may be accounted for by the fact that the rate of heat evolution in the furnace was always about the same, even though the rate of heat transfer from the furnace to the circulating air was slightly less with low air deliveries than it was when higher air deliveries were used. The increase in the range of the bonnet air temperatures accompanying low air deliveries was of the order of only 10 F, and hence was not sufficient to result in any increases in basement temperature or fuel consumption, or in any temperature unbalance in the rooms.

The curve designated as calculated register temperature represents the register air temperatures that would have been used in the design of different forced-air systems for the Research Residence, based on the calculated heat losses for the Residence and the various air deliveries shown in Fig. 1. In order to offset the heat loss from the rooms, either a large air delivery accompanied by low register air temperature or a small air delivery accompanied by a high register air temperature would be used. In this case, in which the Residence was insulated, air deliveries of 1675 cfm, 1300 cfm, and 800 cfm were accompanied by design register air temperatures of 94 F, 103 F, and 131 F, respectively, corresponding to calculated mean bonnet air temperatures of 102 F, 115 F, and 151 F. For the tests shown in Fig. 1 the setting of the bonnet thermostat was not changed, and hence the actual mean bonnet air temperatures were not adjusted to conform to the lower calculated bonnet air temperatures when large air deliveries were used. With the greater air

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deliveries, the calculated mean bonnet air temperatures were lower than the actual, and hence the total time of fan operation was not as great as it would have been if design conditions had been met. In this respect the results

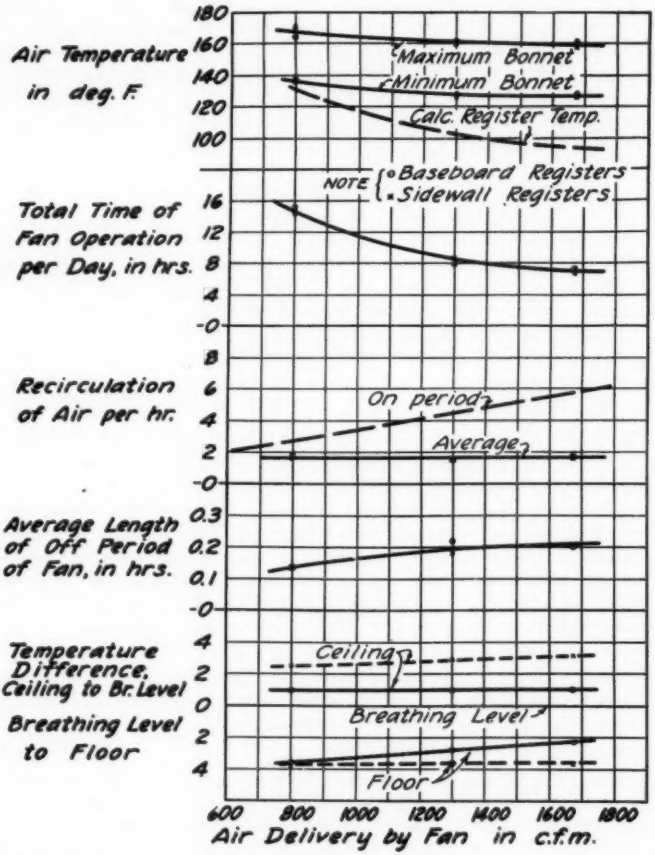

FIG. 1. PERFORMANCE OF BASEBOARD AND SIDEWALL REGISTERS WITH VARIOUS AIR DELIVERIES

Temperature difference of 55 F indoor to outdoor. Bonnet setting 125 F. Stoker-fired furnace

tended to favor to some extent the operation with low air deliveries. However, satisfactory operation was obtained with an air delivery of 1675 cfm, indicating the flexibility of the plant even under conditions when the operating air temperature deviated from the design air temperature.

The total time of fan operation per day, as indicated in the second set of

curves from the top of Fig. 1, was approximately twice as great for 800 cfm delivery as it was for 1675 cfm delivery. However, the electrical input to the fan, expressed in terms of watt-hours per day, was slightly less for the lower air delivery. Hence reducing the speed of the fan resulted in an increase in operating time with no increase in electrical input.

The circulation of 1675 cfm during the *on-period* operation of the fan was equivalent to 5.73 recirculations per hour. If no circulation was assumed to occur during the *off-periods* of the fan, a weighted average value for both *on-periods* and *off-periods* of 1.75 recirculations per hour was obtained. Corresponding values for 800 cfm air delivery were 2.74 recirculations per hour for *on-period* fan operation alone and 1.75 recirculations per hour for the weighted average of *on-periods* and *off-periods*. Hence, the average air recirculation per hour remained constant over a very wide range of air deliveries by the fan, while the *on-period* delivery increased from 2.74 to 5.73 recirculations per hour.

Comfort conditions in the room are improved by any decrease either in the length of *off-periods* of the fan or in the temperature difference between the floor and the breathing level. The lower two sets of curves in Fig. 1 show that an air delivery of 800 cfm resulted in shorter *off-periods* for the fan, but in a slightly larger temperature differential than those occurring with an air delivery of 1675 cfm. The temperature differences shown are the averages for 10 rooms in the Residence, including the relatively large differences obtained in the sun room and the two dormitories. If the open areas of the baseboard registers had been reduced as the air deliveries were lowered, so that a constant register velocity was maintained irrespective of the fan delivery, it is possible that the floor temperatures would not have decreased to the extent shown. In any case, the differences obtained in the length of *off-periods* and in the temperature differential were not of great significance.

Variations in Air Volume Delivered from High Sidewall Registers (*Series* 1, 5, 7-40): The points indicated by crosses and the broken line curves in Fig. 1 show the results obtained with high sidewall registers. The points shown on the four upper sets of curves indicate that no distinction could be made between the results obtained with the baseboard and the high sidewall registers. With both register locations, lower air deliveries resulted in longer time of fan operation per day, the same average number of air recirculations per hour, and in shorter lengths of *off-periods* of the fan as compared with the corresponding results for higher air deliveries.

The lower set of curves indicates that with the high sidewall registers the floor temperature was not affected by the quantity of air delivered when the velocity of the air leaving the registers remained constant at approximately 600 fpm. A comparison of the floor and ceiling temperatures obtained with the two register locations shows that the floor was about 1.0 F warmer and the ceiling about 2.0 F cooler when the baseboard registers were used than they were with the high sidewall registers. The differences are not material. If large sized registers are used in connection with low air volumes, resulting in low register air velocities, higher ceiling temperatures than those shown in Fig. 1, without material changes in floor temperature might be anticipated. This is further discussed in the section on *Changes in Bonnet Air Temperature*. In this connection it is worthy of note that the use of high sidewall registers

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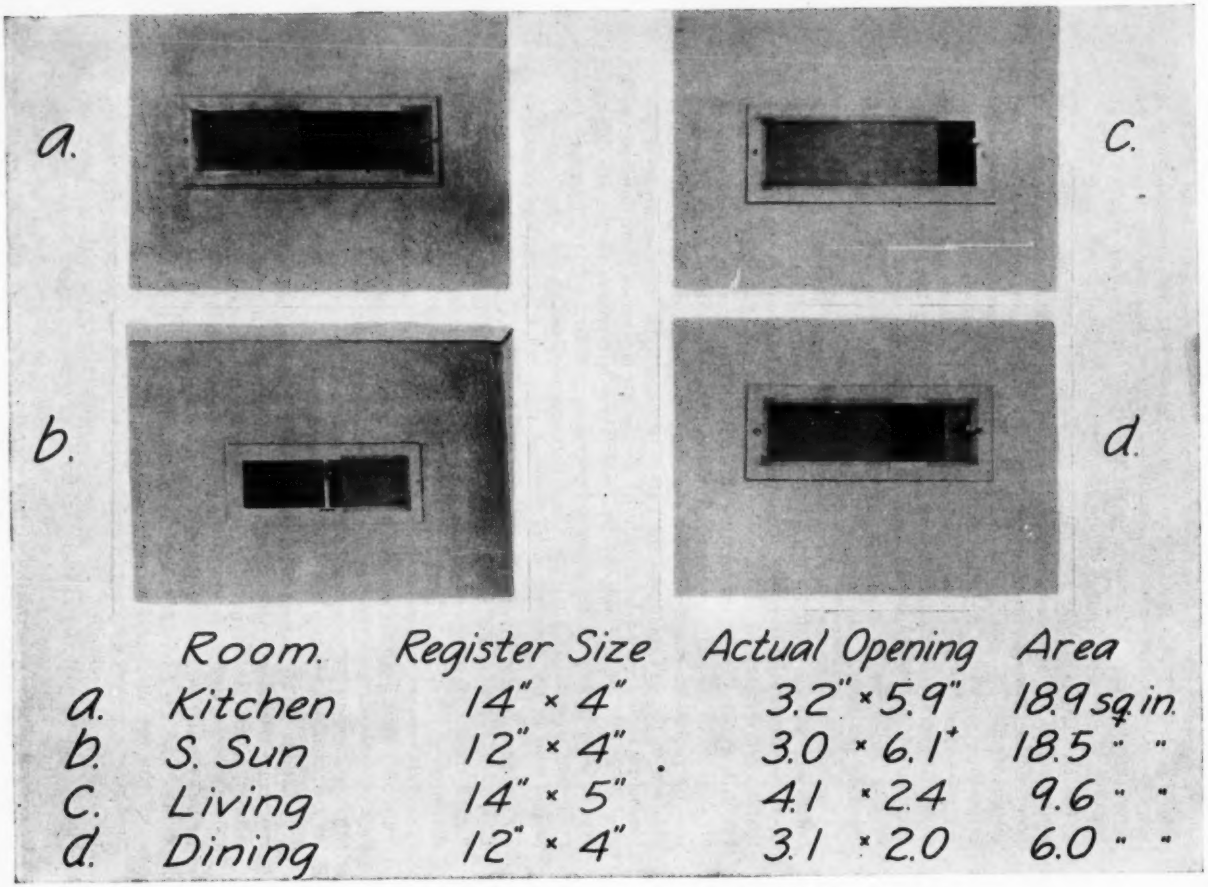

a.
b.
c.
d.
Room. Register Size Actual Opening Area
a. Kitchen 14" × 4" 3.2" × 5.9" 18.9 sq in.
b. S. Sun 12" × 4" 3.0 × 6.1+ 18.5 " "
C. Living 14" × 5" 4.1 × 2.4 9.6 " "
d. Dining 12" × 4" 3.1 × 2.0 6.0 " "

FIG. 2. TYPICAL REGISTER OPENINGS USED WITH LOW AIR VOLUME DELIVERY

permits the employment of high register velocities and low register air temperatures without resulting drafts in the living zone, and the use of small registers that do not interfere with furniture placement.

Based on the results shown in Fig. 1 for both the baseboard and high sidewall registers it may be concluded that the number of air recirculations per hour had no immediate bearing upon either the successful operation of the plant or the comfort conditions produced when the bonnet thermostat was set to give a constant register air temperature of approximately 120 F. No data were obtained to demonstrate the result of operating the plant with a lower setting of the bonnet thermostat to obtain more nearly continuous fan operation at higher air deliveries, but under these conditions only slightly lower temperatures at the ceiling and higher temperatures at the floor level could be anticipated. The results shown in Fig. 1 further indicate that, under the usual conditions of design for intermittently operated plants and with reasonably good building construction, a register air temperature of between 110 and 140 F may be selected and the required air volume be determined from this temperature and the heat losses from the structure with reasonable assurance that the plant will operate satisfactorily. It is of course possible that in the case of poor construction in a severe climate a register air temperature greater than 140 F must be used in order not to exceed seven or eight air recirculations with the resulting difficulties from drafts.

The only feasible argument for the continued use of the arbitrary rule *that a minimum of five or six air recirculations per hour must be provided* is that in some measure it tends to compensate for the general tendency to underestimate the pressure losses in the duct system and through the furnace casing. In other words in the case of a design based on a number of air recirculations per hour as small as three, the reduced air delivery resulting from greatly underestimated pressure losses would probably be more serious than a similar reduction brought about by underestimated pressure losses in the case of a system designed with a comparatively large number of air recirculations. Rules of this nature are but temporary expedients since a deliberate overestimation of air volumes is not the best solution for an underestimation of pressure losses. The solution of the problem lies first in establishing a uniform method of rating furnace-fan units such that the air delivery of the unit is stated in connection with the external pressure loss of the connected duct system, and second in making accurate calculations of the pressure losses in the duct system. From the results obtained it is also apparent that the use of a large sized fan operated at slow speeds will not only provide quiet operation but will also result in short *off-periods*, and comparatively long time of operation with no increase in electrical input. Fundamentally, the success of the two-speed fan in the field has been due to the application of this principle.

Small Warm-Air Registers at High Sidewall Location: Small air volumes accompanied by high register velocities result in small sized register openings. By calculation it is evident that 800 cfm delivery accompanied by 600 fpm register velocity requires a total register opening of $\frac{800}{600}$ or 1.33 sq ft, which is equivalent to a square opening 13.9 in. on each side. Photographs of four register openings actually used in the Research Residence for the series of

FIG. 2. TYPICAL REGISTER OPENINGS USED WITH LOW AIR VOLUME DELIVERY

tests in which 800 cfm of air was circulated are shown in Fig. 2. In many cases the size of these openings was noticeably small. The calculated and actual register openings for the warm-air registers are presented in Table 3. Of the 12 registers used, 6 had actual openings of less than 15 sq in. in area, and the remainder had openings of between 18 and 25 sq in. in area.

A comparison of the calculated and actual register openings, as listed in Table 3, shows that the greatest discrepancies were obtained in the cases of the living room and bathroom. In the latter case, a considerable quantity of warm air was escaping from the bathroom into the central hall, with the result that it was necessary to supply more air than that required to heat the bathroom. In the case of the living room the actual opening was only

TABLE 3—REGISTER OPENING AREAS
(*Actual and Calculated, for 800 cfm Air Delivery*)

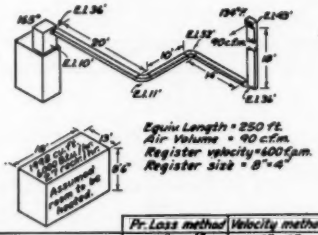
ROOM	CFM CALCULATED	REGISTER AREAS, SQUARE INCHES		
		Calculated	Actual	Difference
Living	144	34.6	9.6	−25.0
Hall	48	11.5	8.2	− 3.3
Dining	36	8.6	6.0	− 2.6
S. Sunroom	82	19.7	18.5	− 1.2
N. Sunroom	82	19.7	25.2	+ 5.5
Kitchen	58	13.8	18.9	+ 5.1
E. Bedroom	102	24.5	18.8	− 5.7
S. W. Bedroom	54	13.0	8.7	− 4.3
Bath	13	3.2	24.5	+21.3
N. W. Bedroom	69	16.6	22.8	+ 6.2
E. Dormitory	62	14.9	12.9	− 2.0
W. Dormitory	50	12.0	12.9	+ 0.9
Total	800	192.1	187.0	= 1.3 sq ft

about 28 per cent of the calculated requirement, indicating either that the heat losses were greatly overestimated or that the room was receiving heat from sources other than that supplied by the air leaving the warm air register. The latter is the most probable explanation since the room was located over the furnace and the smoke pipe. In addition, four of the studding spaces on the inside wall contained warm air stacks, and the outside wall had an adjacent warm chimney. The average temperature shown by two thermometers located in the joist spaces under the living room was about 77 F, whereas the air temperature in the center of the basement at the 60-in. level was only 66 F. For all practical purposes, the living room was heated by a panel heating system consisting of a warm floor and a warm wall. The southwest and east bedrooms were also exposed to two warm studding spaces, whereas the kitchen and sun room were exposed to relatively cool sub-floor temperatures, accounting to some extent for the discrepacies shown in Table 3.

The use of small registers in the case of well insulated structures does not necessarily imply that the stack and branch ducts can be installed with cross-sectional areas as small as those of the register openings. In the example

shown in Fig. 3 it is assumed that the heat loss for a room is 6,000 Btu per hour and the connected duct has an actual length of 62 ft and an equivalent length of 250 ft. Under these conditions the register size, based on a delivery of 90 cfm and a register velocity of 600 fpm, would be 8 in. x 4 in. If it is assumed that the total loss in pressure for the entire warm air duct is to be 0.10 in. and the pressure loss method of calculation as given in the Technical Code of the *National Warm Air Heating and Air Conditioning Association* is used, the stack and duct sizes would be 14 in. x 3¼ in. and 6 in. x 8 in. respectively, both of which are practically fifty per cent greater in cross-sectional area than the 8-in. x 4-in. register.

If the velocity method of duct design is used, and a duct velocity of 600 fpm is assumed, the duct and stack sizes would be only 8 in. x 3 in., but the



	Pr. Loss method	Velocity method
Stack size	14" × 3¼"	8" × 3"
Stack velocity	280 f.p.m.	600 f.p.m.
Duct size	6" × 8"	8" × 3"
Duct velocity	270 f.p.m.	600 f.p.m.
W.A. Duct loss	0.10 in.	0.42 in.
Return duct loss	0.10 in.	0.10 in. +
Furnace press.	0.15 in.	0.15 in.
Total S.P. on fan	0.35 in.	0.67 in. +

FIG. 3. EXAMPLE SHOWING PRESSURES IN HIGH VELOCITY DUCT SYSTEM

total loss in pressure on the warm-air side alone would be approximately 0.42 in., or 4.2 times that based on the pressure loss method. The minimum pressure loss for the entire duct system, including losses in furnace casing and return duct, would therefore be not less than, and probably greater than, 0.67 in. for the velocity method of design, and only about 0.35 in. for the pressure loss method. The fans required in the two cases would be radically different in size and performance characteristics. In fact, a fan required to deliver 800 cfm against a static pressure in excess of 0.67 in. would probably be far different from the fans now in common use for forced-air furnace units. In general, therefore, although the use of small duct systems in connection with low volume, high velocity registers is feasible, such systems must be properly designed with adequate attention given to accurate calculation of the pressure losses. Arbitrary methods of duct design such as those represented by the velocity method should not be used unless the pressure losses in the duct system are also accurately calculated, or unless fans having the necessary characteristics are provided.

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Changes in Bonnet Air Temperatures (*Series* 4, 5, 6-40): The significant results obtained from the series of tests made with three different settings of the bonnet thermostat are shown in Fig. 4. The average bonnet air tem-

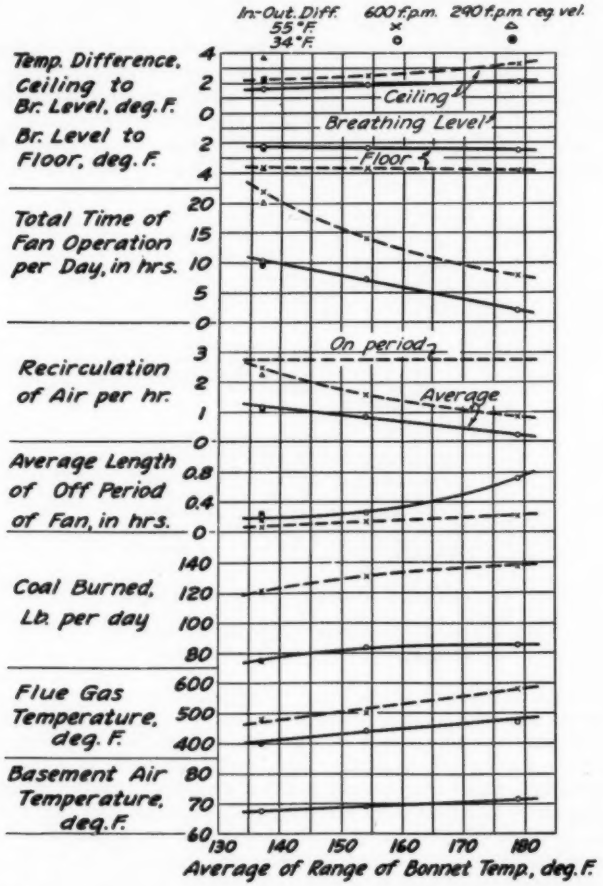
In-Out. Diff. 55°F. 34°F.
600 f.p.m.
290 f.p.m. reg. vel.
Temp. Difference, Ceiling to Br. Level, deg. F.
Br. Level to Floor, deg. F.
Ceiling
Breathing Level
Floor
Total Time of Fan Operation per Day, in hrs.
Recirculation of Air per hr.
On period
Average
Average Length of Off Period of Fan, in hrs.
Coal Burned, Lb. per day
Flue Gas Temperature, deg. F.
Basement Air Temperature, deg. F.
Average of Range of Bonnet Temp., deg. F.

FIG. 4. PERFORMANCE OF SIDEWALL REGISTERS WITH VARIOUS BONNET AIR TEMPERATURES

800 cfm delivery, stoker-fired furnace

peratures shown by the abscissae represent the mean of corresponding maximum and minimum bonnet air temperatures. All tests were made with high sidewall registers, with a total air delivery of 800 cfm, and with register openings as shown in Table 3. With the exception of one case, the register

air velocity was approximately 600 fpm. The full line and broken line curves represent data for indoor-outdoor temperature differences of 34 F and 55 F respectively. The temperature differences between ceiling and breathing level, and between breathing level and floor are shown by the top set of curves. Reducing the bonnet temperature by lowering the setting of the bonnet thermostat tended to slightly lower the ceiling temperatures but had no appreciable effect on the floor temperatures. No data were obtained with different bonnet air temperatures used in connection with higher air deliveries, but Fig. 4 indicates that only slight increases in temperature of the floor and decreases in temperature at ceiling could have been made by operating the plant with lower bonnet air temperatures accompanying the higher air deliveries.

One series of tests (3-40) was conducted at the low air delivery of 800 cfm for which the original size of the register was retained, thus resulting in a register air velocity of approximately 290 fpm instead of 600 fpm. The results are shown by the triangles and solid circles in Fig. 4. The only noticeable effect with the comparatively low bonnet thermostat setting was an increase in the temperature of the air at the ceiling.

Although the effect of reduced setting of the bonnet thermostat on the room temperature gradients was small, in other respects the most satisfactory operation was obtained with the lower values of the bonnet air temperature.

As shown by Fig. 4, reducing the bonnet temperature resulted in a longer total time of fan operation per day, a larger average air recirculation, and much shorter *off-periods*. In fact the setting of the bonnet thermostat can be so reduced as to produce practically continuous fan operation even on days in which the outdoor temperature is not extreme. With such reduced settings of the upper limit of the bonnet thermostat, above which temperature the fan would operate, if the limit thermostat for the stoker or burner were set at a temperature high enough to take care of extreme weather conditions, the operating bonnet air temperature would be automatically proportioned to the heating demands of the structure. That is, in very cold weather the operating bonnet air temperature would automatically rise higher than the temperature indicated by the upper limit of the bonnet thermostat setting, with the net result that the heat output would be adequate. On the other hand, if the bonnet thermostat setting had been increased to a very high value, the plant would have tended to operate entirely under gravity air circulation. With a high setting of the bonnet thermostat and a given adjustment of the duct dampers the temperature balance between rooms over a wide range of outdoor temperatures was not as satisfactory as that obtained with a similar adjustment in the case of lower settings of the bonnet thermostat. Higher bonnet settings, particularly for a solid fuel plant in which some heat liberation occurs during the *off-periods* of the stoker, result to some extent in uncontrolled gravity circulation of the air during the *off-periods* of the fan and thus tend to produce inequalities in the temperature distribution between rooms.

As shown by Fig. 4, the higher settings of the bonnet thermostat were also accompanied by greater coal consumption on a given day. The increase in coal consumption could be largely accounted for by an increase in the heat losses from the basement resulting from higher basement temperature and by an increase in the flue losses accompanying higher flue gas temperatures. The latter was particularly true during the *off-periods* of the stoker.

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From the standpoint of low temperature difference from floor to ceiling, long total time of operation of the fan, shorter lengths of *off-periods*, and lower fuel consumption, the most favorable results were obtained with the lower settings of the bonnet thermostat. In this connection it should be noted that when lower bonnet air temperatures are used the actual operation of the plant more nearly approaches that with continuous operation of the fan, thus approximating the conditions assumed for design. Furthermore, the amount of dampering required in the duct system to take care of variations in outdoor temperature will be dependent upon the range of bonnet air temperatures maintained and should be the least with low ranges of bonnet air temperatures. Lower bonnet air temperatures can be used in conjunction with high sidewall or ceiling registers than with baseboard registers. The lowest allowable limit of the bonnet air temperature is dependent upon the lowest register air temperature tolerable without resulting in objectionable drafts in the rooms.

SUMMARY

The results of these tests may be summarized as follows:

1. The forced-air heating system proved to be very flexible insofar as its adequacy to heat the Residence with air volumes and register air temperatures deviating from those selected for the design of the plant was concerned.
2. At a bonnet air temperature of approximately 150 F and with a register air velocity of approximately 600 fpm, the variations in the fan delivery equivalent to from three to six air recirculations per hour had no immediate bearing upon either the successful operation of the plant or the comfort conditions produced.
3. With intermittent operation of the heating plant at an approximately constant bonnet air temperature, although the fan delivery varied from three to six air recirculations per hour, the average air recirculations per hour including both the *on-periods* and *off-periods* remained constant.
4. The use of baseboard registers resulted in floor temperatures approximately 1.0 F warmer and ceiling temperatures approximately 2.0 F cooler than those obtained with high sidewall registers. The use of high sidewall registers permitted the employment of high register velocities and low register air temperatures without resulting drafts in the living zone.
5. The use of high sidewall registers in connection with low air volumes and high register air velocities permitted the employment of registers smaller than those usually observed in common practice.
6. The use of an excessively small duct system in connection with small registers cannot be justified unless accurate calculations of the anticipated pressure losses are made.
7. From the standpoint of low temperature difference between floor to ceiling, long total time of operation of the fan, shorter lengths of *off-periods*, and lower fuel consumption, the most favorable results were obtained with the lower settings of the bonnet thermostat.

ACKNOWLEDGMENT

The results presented in this paper were obtained in connection with the investigation of warm-air furnaces and heating systems in the Research Residence[3] at the University of Illinois, conducted by the Engineering Experi-

[3] The Research Residence in Urbana, Illinois, was built, furnished, and completely equipped specifically for research work in warm-air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

ment Station. This investigation is a cooperative project sponsored jointly by the *National Warm Air Heating and Air Conditioning Association* and the Engineering Experiment Station. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is made to D. W. Thomson and H. R. Stover, formerly Research Graduate Assistants, for their services in conducting the tests and in the reduction of the test data. Acknowledgment is also made to the various manufacturers who cooperated by furnishing instruments and equipment.

DISCUSSION

E. K. CAMPBELL, Kansas City, Mo.: This is a very timely paper. It cannot be stressed too often or too much that air volume should be proportioned to the Btu load required to produce comfort conditions within the building or room.

While this paper is prepared entirely from the standpoint of residence heating, the things that are pointed out have their application in all types and kinds of buildings.

For instance, the introduction of the air at the floor producing three degrees lower temperature differential is comparable to the modern practice with unit heaters, discharging the air downward so that it strikes the floor and mixes with the cold air on the floor, thereby producing a large induced circulation and raising the temperature at the floor and reducing the temperature at the ceiling.

In audience room work it is not desirable to stop the fan at any time the audience is in the room, while in the residence work intermittent operation of the fan, according to the temperature of the room, is desirable. The fact that the air movement determines the temperature differential means that in a high ceiling room if the fan is stopped, immediately the air which is cooled by the walls and windows drops to the floor and tends to lower the temperature in the lower strata of air which are the ones which determine the comfort of the audience. Also if the room is a little warm it is necessary to maintain motion next to the body in order to carry away the body heat as rapidly as possible. So the fan must operate constantly while the audience is in the room and the air must be introduced in such a way that if it does come in cool it does not cause an objectionable draft anywhere within the room. With these things in mind it is then necessary not only to determine the volume of air according to the Btu load, but it must not be introduced at too high a temperature. In our practice we have found it advisable to restrict the temperature rise of the air to approximately 40 to 45 degrees, with 55 deg temperature rise as the air passes over the heating surface as the maximum for good practice. That means that each cubic foot of air absorbs from two-thirds to one Btu as it passes over the heating surface.

Temperature rise of less than 40 deg can be used successfully in very large rooms where the air can be introduced high enough to permit thorough mixture with the other air so as to prevent objectionable drafts.

Any paper that spreads the gospel of large volumes at lower temperatures and reasonable size ducts instead of the extremely small ones that we sometimes see, helps to advance the art of heating by air movement.

JOHN EVERETTS, JR., Pittsburgh, Pa.: I notice that the authors state that register temperatures as low as 94 F and 103 F might be used with the larger quantities of air. It has been my experience that such low register temperatures always created an uncomfortable condition and that this discomfort was only alleviated after the register temperatures had been raised to at least 110 F. As the authors do not show in the paper any actual test with such low register temperatures, I would like to ask if they actually tried these temperatures and what comfort conditions they experienced.

W. L. FLEISHER, New York, N. Y.: Were measurements taken of the relative humidity prevailing at the same time dry-bulb readings were taken of the conditions prevailing in the house?

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PROFESSOR KRATZ: These readings had not been reported, but the relative humidity was about 25 to 35 per cent for indoor-outdoor temperature differences of 34 F. In any event the relative humidity was always practically the same for any given weather, and hence did not affect the performance of the plant. These studies were confined to the effect of plant performance on temperature gradients, etc., and the effect of relative humidity on comfort would be the same, irrespective of changes in the plant. Mr. Fleisher was critical of the fact that a paper presented to the Society as indicating the performance of a heating system should not have included the relative humidity as an important function of the maintenance of comfort. Studies are being made to determine just what effect the moisture content has on the requirements for heating, and what the psychological reaction on the occupants is with higher or lower relative humidities. The 25 to 35 per cent relative humidity which Professor Kratz thought prevailed during these tests was a relative humidity which in Mr. Fleisher's opinion was a reasonable moisture content to prevent uncomfortable dryness. He thought that the authors should include reference to relative humidity, give some conclusions drawn from the relative humidity prevailing, or make a test to find what would have been required in the way of fuel consumption with different amounts of moisture in the enclosure before the paper appears in the Society's TRANSACTIONS.

Mr. Everetts questions the feasibility of operating with register air temperatures below 110 F. Later tests than those reported in the paper have proved that with high sidewall registers it is possible to obtain satisfactory comfort conditions with register air temperatures as low as 95 F. With baseboard registers, however, the authors are inclined to agree with Mr. Everetts that temperatures below 110 F may be questionable, unless deflecting type registers, such as were used in the Residence are employed.

No. 1216

EFFECT OF VANES IN REDUCING PRESSURE LOSS IN ELBOWS IN 7-INCH SQUARE VENTILATING DUCT

By M. C. STUART,* C. F. WARNER,** AND W. C. ROBERTS †
BETHLEHEM, PA.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with Lehigh University.

INTRODUCTION

THIS is the second of a series of papers dealing with pressure loss caused by elbows in ventilating ducts. The first paper, presented by the authors (17) †† described the method of measuring pressure loss, developed a method of expressing pressure loss of an elbow as equivalent length of duct, and reported upon pressure losses in elbows without vanes in 8-in. round ventilating duct. Reference should be made to this previous paper for a description of the apparatus and method of investigation used.

The present paper presents the results of an investigation of elbows of one aspect ratio in 7 x 7-in. square duct, and in particular studies the effect of various forms of vanes in reducing the pressure loss in the elbow.

PRESSURE LOSS IN RADIUS ELBOWS WITHOUT VANES

Apparatus for the study of the pressure drop in a single elbow consisted of a fan, discharge box, 30 ft of approach duct, the elbow under investigation, 105 ft of discharge duct, plenum chamber, and measuring nozzle. The duct was made up of standard 5 ft sections fastened by the Pittsburgh lock which was taped to prevent air leakage. Elbows of various radius ratios were tested

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† Instructor in Mechanical Engineering, Lehigh University.
†† Numbers in parentheses refer to Bibliography.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, St. Paul, Minn., June, 1942.

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at velocities ranging from 400 to 4000 fpm. The radius ratio of an elbow is defined by the equation:

$$\frac{R}{W} = 0.5 \frac{1 + \frac{R_1}{R_0}}{1 - \frac{R_1}{R_0}}$$

in which

R = mean radius of elbow
W = width of elbow in plane of bend
R_0 = outside radius
R_1 = inside radius.

The standard elbows (without vanes) tested in this part of the investigation are shown in Fig. 1. Table 1 gives the dimensions of the elbows shown in Fig. 1.

Curve A of Fig. 2 indicates that as the radius increases, the additional equivalent length (corresponding to the pressure loss) decreases until a constant value of equivalent length is reached at a value of 28 in. or greater inside radius. The additional equivalent length is defined as:

$$\frac{\text{Measured pressure drop of a section of duct including an elbow minus the calculated pressure drop of a straight section of duct having the same centerline length.}}{\text{The loss per foot of straight duct.}}$$

The total equivalent length is defined as the additional equivalent length plus the centerline length of elbow. (For pressure loss per foot of duct see ASHVE HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1942.)

As in the case of elbows in round duct, the pressure loss, expressed in total equivalent length, was found to be independent of velocity. *Curve B* indicates that total equivalent length decreases with increased radii until a minimum value of 9.5 ft of duct is reached, whereupon any further increase in radius will cause an increase in total equivalent length. These results are also given in tabular form in Table 1.

The elbow with a radius ratio of 2.6 has the minimum loss expressed as total equivalent length. This result is substantiated by data by Kirkwood, (1), replotted by Konzo, (10).

ELBOWS WITH SQUARE OUTSIDE CORNERS

In an earlier investigation by Wirt, (7) it was found that elbows with square outside corners have less loss than those with well rounded outside corners. This result which seemed surprising was upheld by data and pictures presented in the paper. However, two later investigations, one by Brown, (2), and the other by Madison and Parker, (5), presented data that showed the loss for an elbow with square corner to be larger than for one with a rounded corner. In order to throw light on this question, four elbows, with square outside corners shown in Fig. 3, were tested. All of these elbows

were tested over a velocity range of 400 to 3500 fpm. The results, as shown by *Curve C*, Fig. 2, show that in all cases the square outside elbows have a greater loss than the corresponding radius elbows. The excess loss becomes much more pronounced as the inside radius increases.

ELBOWS IN SERIES

In order to study the effect of elbows in an actual installation, a test system was set up using three elbows and 130 ft of straight duct arranged as follows: the fan and discharge box, 16 ft of duct, the first measuring station, 7 ft of duct, the first elbow, 25 ft of duct, the second elbow, 25 ft of duct, the third elbow, 35 ft of duct, the second measuring station, 15 ft of duct, and the

FIG. 1. STANDARD ELBOWS

plenum chamber. The straight duct between the measuring stations was 92 ft long.

Three different sets of three elbows each, contributing 56, 40, and 31 per cent of the total loss were used. Set No. 1 was made up of three 0.5 radius ratio elbows; set No. 2, three 1.1 radius ratio elbows; and set No. 3, one 0.5, one 1.1, and one 2.6 radius ratio elbow. Tests were run at velocities from 750 to 3500 fpm and in all cases, the calculated drop differed from the measured drop by only $\pm$ 0.02 in. of water for the low velocities and $\pm$ 0.04 in. of water for the high velocities.

PRESSURE LOSS WITH VANES AND SPLITTERS

The present investigation was directed largely to determine the action of different types of vanes and splitters in reducing pressure loss in elbows. Several previous investigators have shown agreement as to the effects of certain shapes of vanes. Brown, (2), Patterson, (8), and Kline, Tupper, and Green, (9), have presented results of separate investigations of different sizes and shapes of vanes, and all have reached the conclusion that the simple splitter bent on a moderately long radius, making 90 deg arc, is the most practical vane. Therefore, this investigation includes only the simple splitter, the Keiber vane, and three commercial vanes.

To avoid confusion, a vane is herein defined as any curved section used to direct the air flow, while a splitter is a thin vane which includes a circular 90 deg arc.

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COMMERCIAL VANES

All vanes may be grouped into three classes: *A*—those made up of a large number of small splitters; *B*—those made up of a small number of large splitters bent on a large radius; *C*—hollow vanes having different outside and inside curvature.

Each of the three commercial vanes tested came under a different classification so that a comparison could be made. All three commercial vanes

TABLE 1—ELBOWS WITH ROUND OUTER RADIUS

ELBOW DESIGNATION	INSIDE RADIUS IN IN.	ADDITIONAL EQUIVALENT LENGTH IN FT	TOTAL EQUIVALENT LENGTH IN FT
Standard	0	39.2	39.7
B	1½	24.6	25.3
C	4	13.2	14.2
D	7	9.5	10.9
E	11	7.7	9.6
F	15	7.1	9.5
G	21	6.4	9.6
H	28½	5.9	10.0
SQUARE OUTER RADIUS			
A	0	40.6	41.1
A–A	1½	22.8	23.5
A–B	7	19.4	20.8
A–C	11	16.8	18.7
COMMERCIAL VANES			
	VANE DESIGNATION		
A	A	21.3	21.8
	B	16.47	17.02
	C	17.4	17.9
A–A	A	16.6	17.3
	B	13.7	14.4
	C	14.3	15.0

were tested in the miter square outside corner elbow and the 1½ in. inside radius square outside corner elbow. The test velocities ranged from 1200 to 3500 fpm.

The results as given by Fig. 4 and Table 1 show that vanes *B* and *C* have the lowest loss when used in each elbow. The use of these vanes in the 1½ in. inside radius elbow reduces the loss only 34 per cent.

Vane *C* was cut down by the manufacturer from a vane for larger sized duct and, consequently, restricted the flow more than would be experienced normally.

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SPLITTERS

In order to obtain information on the characteristics of various splitters and combinations of splitters, the miter inside and square outside elbow, the standard miter elbow, the 1½ in. inside radius elbow, and the 4 in. inside radius elbow, were fitted with various splitters. The splitters having radius ratios of 0.2 to 1.2 were tested in one splitter, two splitter, and three splitter com-

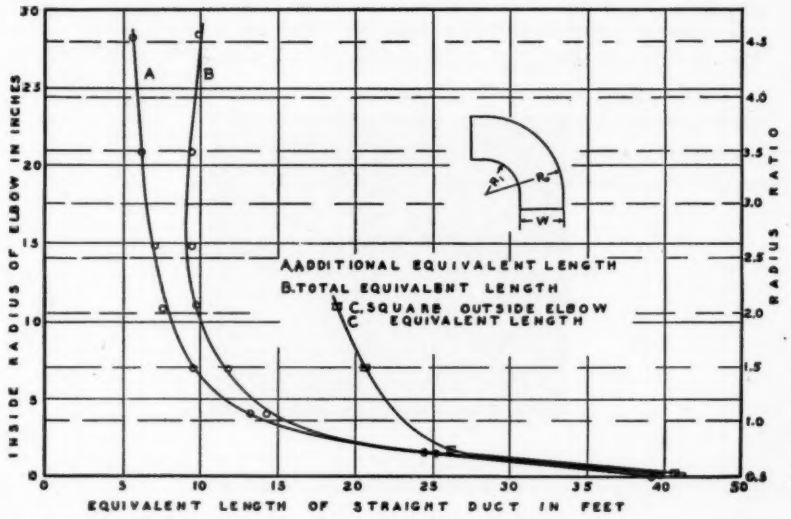

FIG. 2. RELATION BETWEEN INSIDE RADIUS OF ELBOWS AND EQUIVALENT LENGTH OF STRAIGHT DUCT

binations. In all cases the center of curvature of the splitter was either the inside corner of the elbow or the center of curvature of the elbow.

The results as given by Figs. 4 and 5 show that, in each elbow and at all velocities, the single splitter reduces the loss caused by the elbow with increase in radius ratio until a minimum value is reached, then, the loss begins to increase with increased radius ratio. It will be noted that splitters either close to the inside or outside have little effect. The results, also, show that the loss of the elbow can be reduced by using 2 or 3 splitters instead of one. The effect of the splitter becomes less as the inside radius of the elbow increases. In fact, the results of the tests on the 4 in. inside radius elbow show very little reduction in loss by the use of splitters.

Because the splitter with a radius ratio of 0.4 was the best when used in a miter elbow, four of these were tested in combination and a very material reduction in loss was noted. The addition of 2 in. of trailing edge to the splitters reduced the loss a little but not enough to make their use advantageous.

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In order to determine the extent to which the loss could be reduced if the best type of theoretical vane were used, three Keiber vanes, (13), were tested in the miter inside and square outside elbow. The results, Fig. 4, show that the loss is lower than that obtained by any other vane tested.

MODEL STUDY

In order to obtain a picture of the flow pattern in an elbow, a model, (see Appendix, Fig. A), was made for use in the *schlieren* apparatus. The model was made of three sections: the inlet, the elbow, and the discharge. The inlet section was fitted with a bell mouth and an egg crate straightener to

FIG. 3. ELBOWS WITH SQUARE OUTSIDE CORNERS

establish uniform flow at the entrance to the elbow. The elbow was made by fitting different shaped outlines of wood between two ½ in. glass plates held in place by an aluminum frame. The discharge section, fitted with a slide gate to control the velocity, led from the elbow to the suction side of a constant speed fan.

While the 1 x 3-in. duct formed by the model did not accurately reproduce the secondary flow as found in the 7 x 7-in. duct, it reproduced the flow in the plane of the bend to be sufficiently indicative of the true flow for practical consideration.

As the air stream entered the elbow, an electrically heated wire grid heated streams of air ½ in. apart, and these heated streams appeared as white streaks in the *schlieren* picture. (For a description of *schlieren* apparatus, see Appendix.) The *schlieren* apparatus was used for two reasons: the flow streak disappears when the turbulence becomes too great and there is no foreign element introduced into the air stream.

The elbows in the model had radius ratios of 0.5, 0.87, 1.50, and 2.17. The balsa wood splitters and vanes used, having radius ratio of 0.2, 0.4, 0.6, 0.8, 1.0, 1.2, 1.4, and 1.6 were held in place by cementing them to the glass. In the accompanying photographs the elbow walls and vanes have been out-

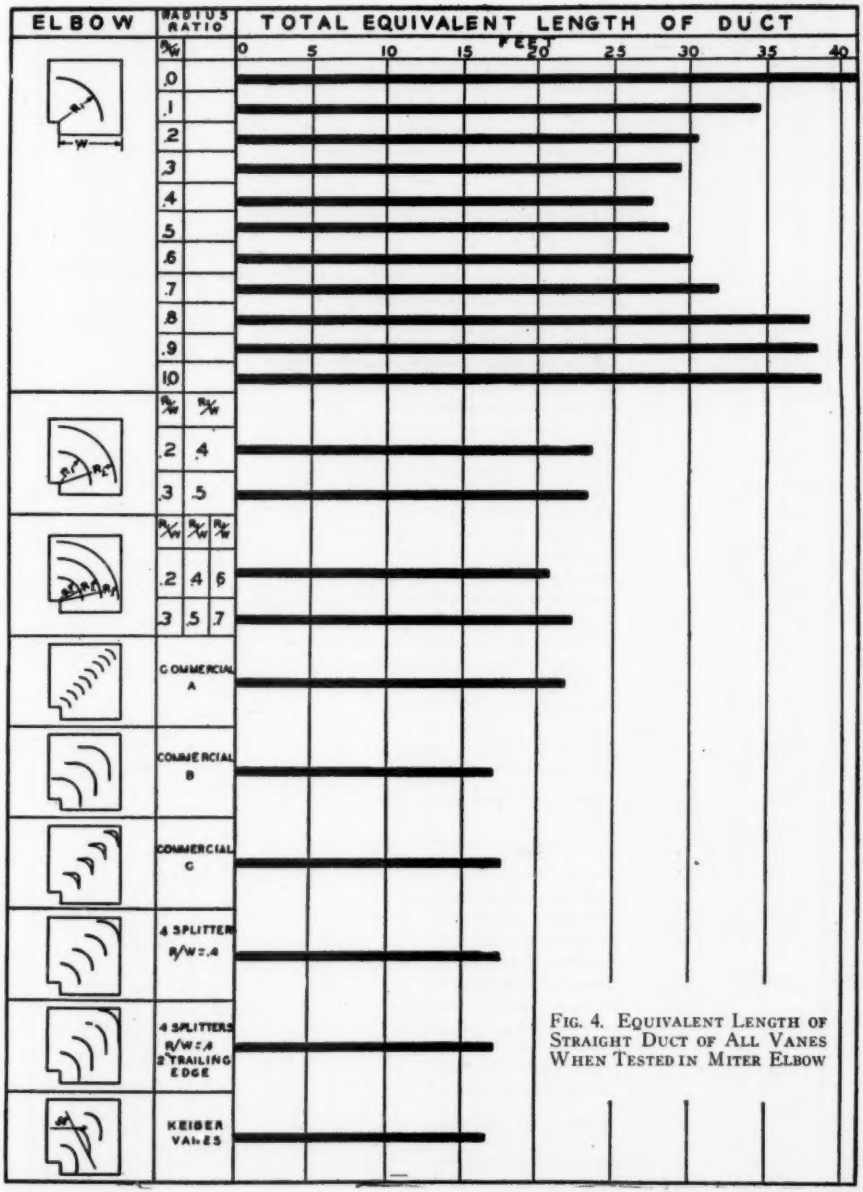

FIG. 4. EQUIVALENT LENGTH OF STRAIGHT DUCT OF ALL VANES WHEN TESTED IN MITER ELBOW

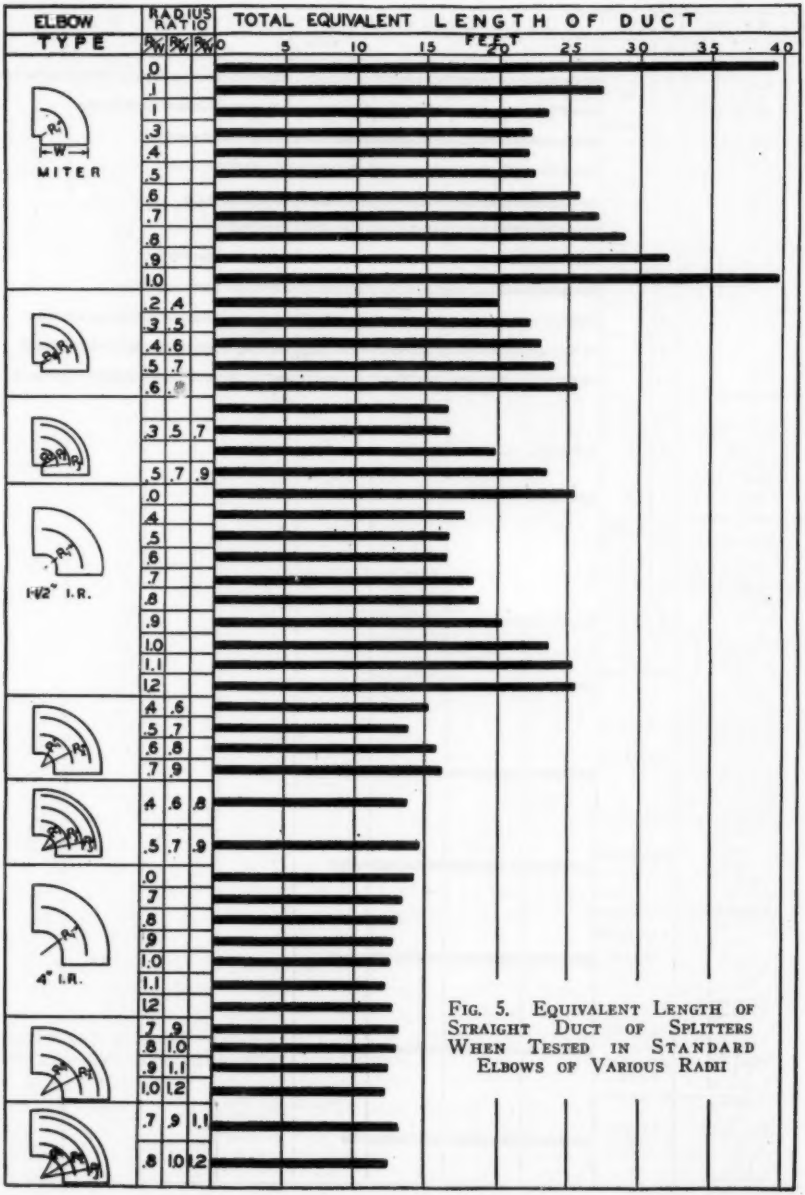

FIG. 5. EQUIVALENT LENGTH OF STRAIGHT DUCT OF SPLITTERS WHEN TESTED IN STANDARD ELBOWS OF VARIOUS RADII

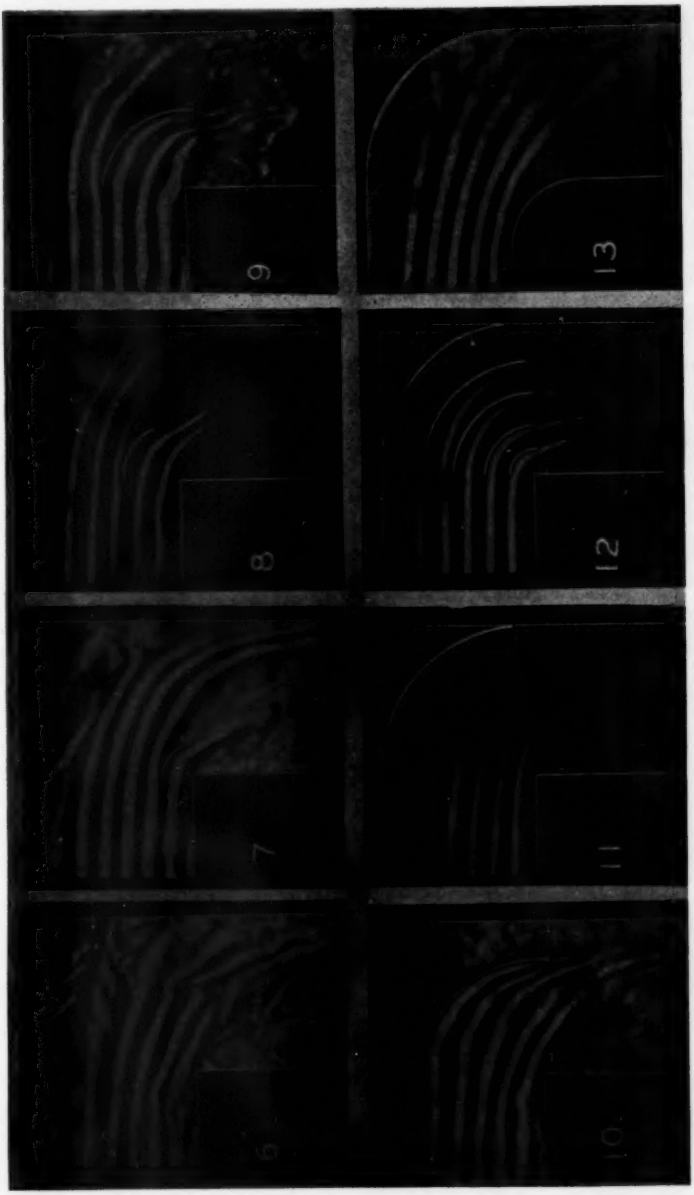

FIG. 6. FLOW IN MITER ELBOW. FIGS. 7-12. FLOW IN MITER ELBOW WITH SPLITTER. (7) *0.2 R/W*; (8) *0.4 R/W*; (9) *0.6 R/W*; (10) *0.8 R/W*; (11) *1.0 R/W*; (12) *All four splitters.* FIG. 13. FLOW IN 0.87 RADIUS RATIO ELBOW.

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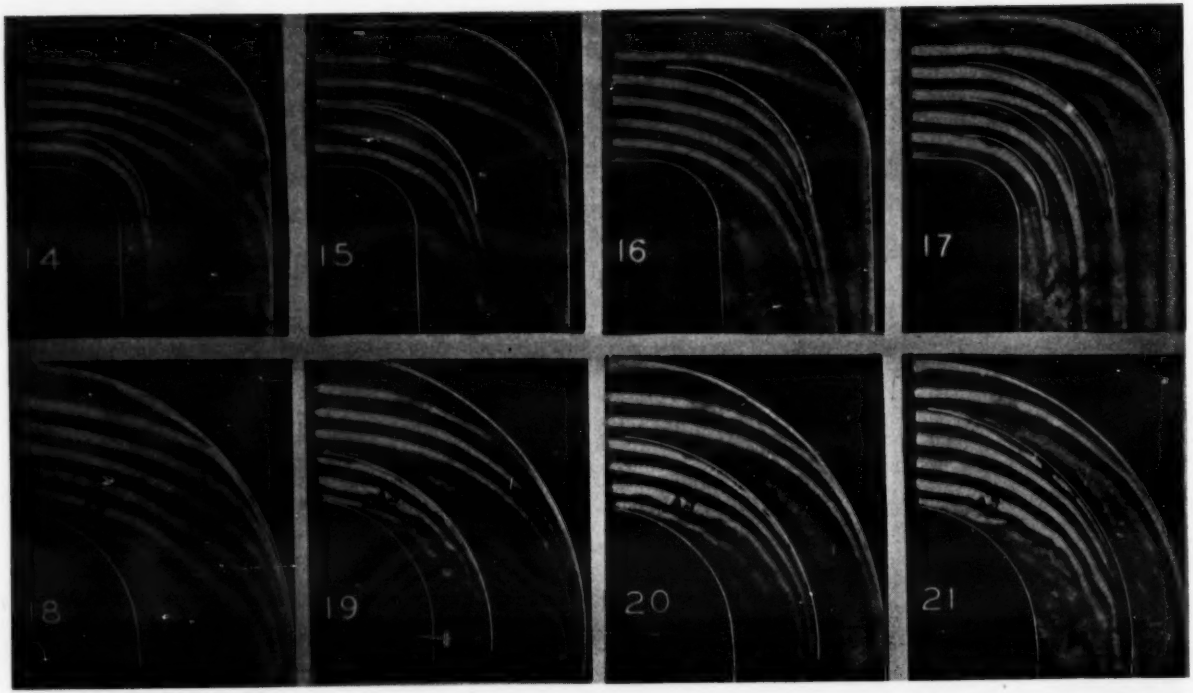
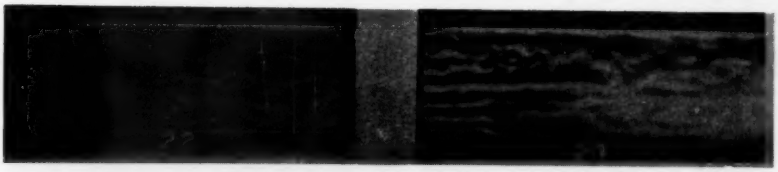


FIGS. 14-17. FLOW IN 0.87 RADIUS RATIO ELBOW WITH SPLITTER. (*14*) *0.6 R/W*; (*15*) *0.8 R/W*; (*16*) *1.0 R/W*; (*17*) *All three splitters*. FIG. 18. FLOW IN 1.50 RADIUS RATIO ELBOW. FIGS. 19-21. FLOW IN 1.50 RADIUS RATIO ELBOW WITH SPLITTER. (*19*) *1.2 R/W*; (*20*) *1.4 R/W*; (*21*) *1.6 R/W*.



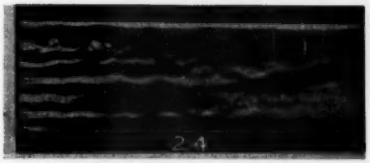


FIG. 22. (*upper left*) FLOW AFTER THE MITER ELBOW.

FIG. 23. (*upper right*) FLOW AFTER THE MITER ELBOW WITH FOUR 0.4 *R/W* SPLITTERS.

FIG. 24. (*left*) FLOW AFTER THE MITER ELBOW WITH FOUR KEIBER VANES.

lined with white ink to make them more prominent. The slightly rough appearance of some of the vanes is due to the cement.

In Figs. 13 to 17 of the 0.87 radius ratio, it will be noted that the walls are not concentric. The inside radius is too large for the outside radius making the separation at the outside wall less than would be experienced in the correct elbow.

The result of this work is given by Figs. 6 to 45. These pictures tell the same story as the loss determination tests. Figs. 6, 13, 18, and 26 show that as the inside radius is increased, the flow becomes smoother and less turbulent. They also indicate that the shape of the inside corner greatly affects the pressure loss. Figs. 31 and 32 show that the square outside elbow does cause much turbulence which disproves the contention of Wirt (*7*). Fig. 35 shows the way in which the turbulence is reduced by rounding the outside of a miter elbow. Figs. 7, 8, 9, 10, and 12 show how each splitter bends the flow into a smooth path until, when all five splitters are used, the air leaves in almost as smooth a flow as when it enters. Figs. 19, 20, 21, and 25 show how little the splitters affect the flow of a large radius elbow. The best flow patterns for the miter elbow are given by Figs. 29 and 30 using the four 0.4 radius ratio splitters and the Keiber vanes.

Figs. 22, 23, 24, and 38 to 45 give a picture of what takes place after the various elbows with and without splitters. These views were taken at a point 2 in. after the elbows. Fig. 37 shows the flow before the elbow and should be used as a standard of excellence. Figs. 23 and 24 show to what extent the flow can be smoothed out by the use of proper vanes in a miter elbow over that of Fig. 22 without any vanes. Fig. 43 of the flow after a large radius elbow shows how little a good elbow disturbs the air flow.

CONCLUSIONS AND RESULTS

From this investigation several significant conclusions may be drawn:

1. The loss caused by an elbow may be reduced by increasing the radius until a minimum value is reached by an elbow having a radius ratio of 2.6.

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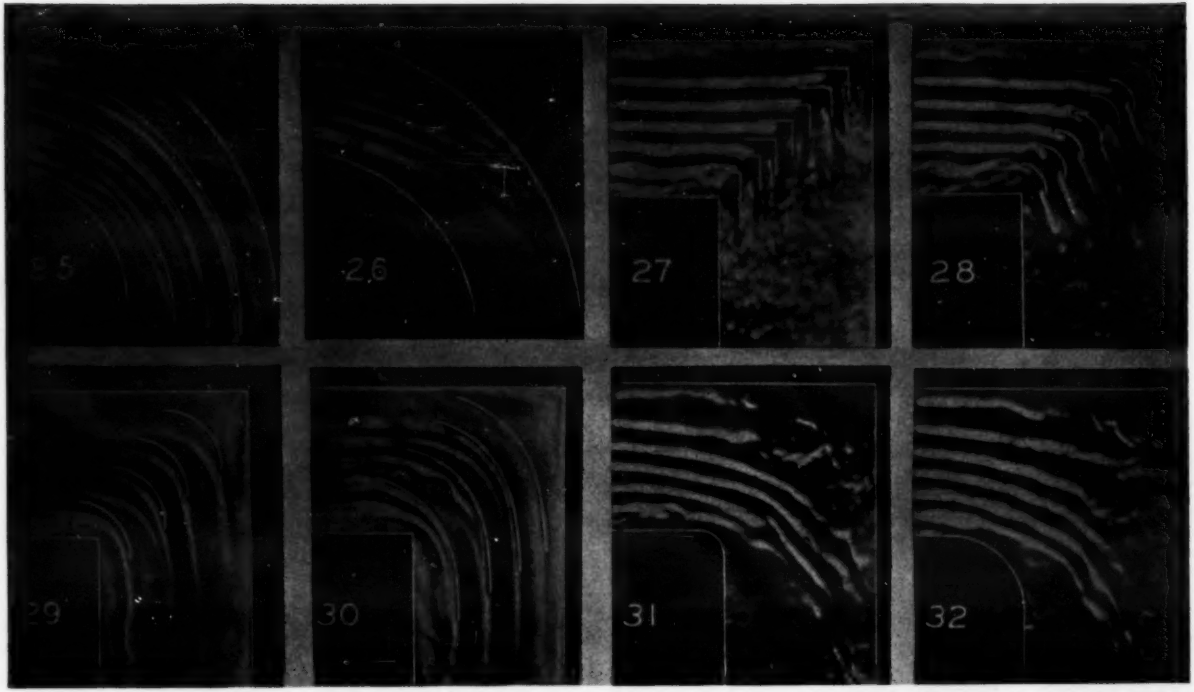
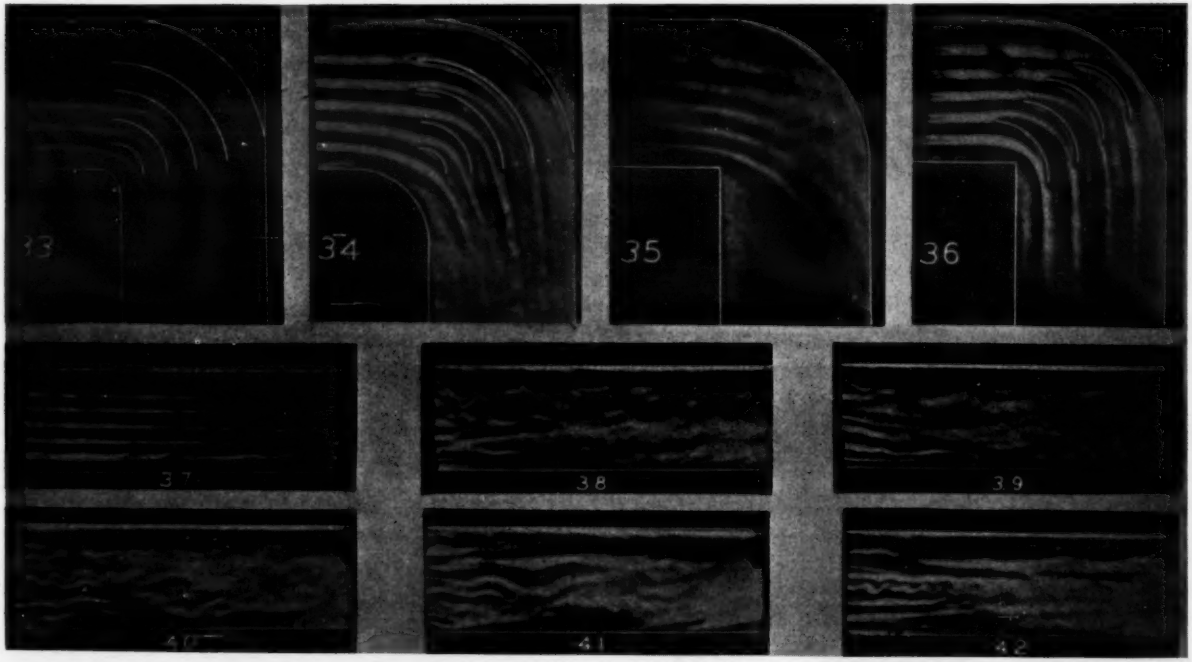
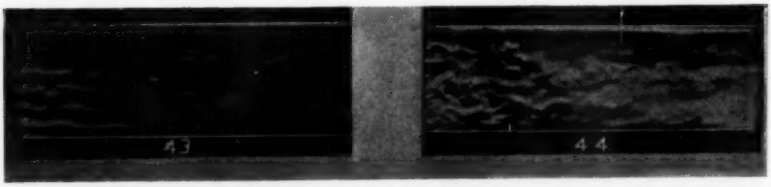


FIG. 25. FLOW IN 1.50 RADIUS RATIO ELBOW WITH ALL OF SPLITTERS. FIG. 26. FLOW IN 2.17 RADIUS RATIO ELBOW. FIGS. 27-30. FLOW IN MITER ELBOW. (*27*) *With right angle vanes;* (*28*) *with small radius splitters;* (*29*) *with four 0.4 R/W splitters;* (*30*) *with four Keiber vanes.* FIGS. 31-32. FLOW IN SQUARE OUTSIDE ELBOW. (*31*) *Small inside radius;* (*32*) *medium inside radius.*



FIGS. 33-34. FLOW IN SQUARE ELBOW. *(33) small inside radius with splitters; (34) medium sized inside radius with splitters.* FIGS. 35-36. FLOW IN STANDARD MITER ELBOW. *(36) With four 0.4 R/W splitters.* FIG. 37. FLOW IN DUCT BEFORE AN ELBOW. FIG. 38. FLOW AFTER STANDARD MITER ELBOW. FIGS. 39-40. FLOW AFTER 0.87 RADIUS RATIO ELBOW. *(40) with one splitter in middle.* FIGS. 41-42. FLOW AFTER 1.50 RADIUS RATIO ELBOW. *(42) with one splitter in middle.*



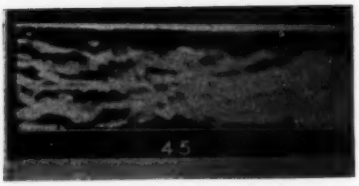


FIG. 43 (*upper left*). FLOW AFTER 2.17 RADIUS RATIO ELBOW.

FIGS. 44-45. FLOW AFTER SQUARE OUTSIDE ELBOW. (*44*) (*upper right*) *small radius inside.* (*45*) (*left*) *medium radius inside.*

2. Elbows with square outside walls cause larger losses than do elbows of curved outside walls.
3. Splitters very materially reduce the loss of miter elbows.
4. The simple 90 deg arc splitter has almost as much effect on the reduction of the loss as do the more complicated vanes.
5. An *easy* elbow has less loss than the miter equipped with the best vanes.
6. Equivalent lengths of pipe for different elbows are presented in Table 1.

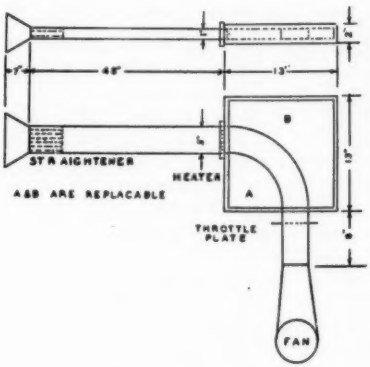
BIBLIOGRAPHY

1. *The Sanitary Age,* December 1931, Karl E. Kirkwood.
2. Friction of Air in Elbows, by A. I. Brown. (*Power Plant Engineering,* Vol. 36, August 15, 1932.)
3. Loss of Pressure Due to Elbows in the Transmission of Air Through Pipes or Ducts, by F. L. Busey. (ASHVE TRANSACTIONS, Vol. 19, 1913.)
4. *Fan Engineering,* Third Edition, pp. 81-105, Buffalo Forge Co.
5. Pressure Losses in Rectangular Elbows, by R. D. Madison and J. R. Parker. (*ASME Transactions,* Vol. 58, p. 167 AER 2 and Discussion of Pressure Loss in Rectangular Elbows.)
6. A Chart Method for Locating Splitters in Elbows, by R. D. Madison. (*American Artisan,* February 1937.)
7. New Data for the Design of Elbows in Duct Systems, by Loring Wirt (*General Electric Review,* Vol. 30, No. 6, June 1927, p. 286.)
8. Corner Losses in Ducts, by G. N. Patterson. (*Aircraft Engineering,* Vol. 9, 1937.)
9. The Design of Corners in Fluid Channels, by G. L. Klein, K. F. Tupper and J. J. Green. (*Canadian Journal of Research,* Vol. 3, 1930.)
10. Winter Air Conditioning (published by *National Warm Air Heating and Air Conditioning Association,* 1939, p. 363.)
11. *Engineering Experiment Station Bulletin No. 52,* University of Illinois, A. P. Kratz, and J. R. Fellows.
12. The Energy Loss Around 90 Deg Bends in Tubes of Uniform Cross-Section, by A. Hofmann. (Miss. Hydr. Inst. Techn. Hochschule, Munchen. Vol. 3, 1929.)
13. Keiber, Ingenieur Archiv. Vol. 3, No. 5, 1932.
14. Energy Loss in Elbows, by H. Kirchbach. (Miss. Hydr. Inst. Techn. Hochschule, Munchen. Vol. 3, 1929.)
15. The Loss of Energy in Duct Elbows and Branches, by M. A. Bouchayer. (Hydro-electric. Kongresz, Grenoble, 1925.)

16. Vane Grids for the Deflection of Fluid Flows with Small Energy Losses, by G. Krober. (Ingenieur-Archiv., Vol. III, December 1932.)
17. ASHVE RESEARCH REPORT No. 1211—Pressure Loss Caused by Elbows in 8-Inch Round Ventilating Duct, by M. C. Stuart, C. F. Warner and W. C. Roberts. (ASHVE TRANSACTIONS, Vol. 48, 1942.)

APPENDIX

The *schlieren* apparatus has been used in the study of flow phenomena by many experimenters. In recent years it has been used most extensively at the National Physics Laboratory in England. While basic arrangement of the apparatus need not be changed, by changing the design of the cut-off or stop and small mirror, many varied effects can be produced which are best



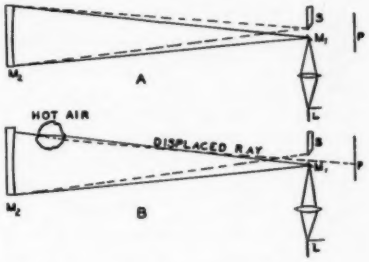


FIG. B. SCHEMATIC DRAWING OF SCHLIEREN APPARATUS

FIG. A. (*left*) ELBOW MODEL USED IN SCHLIEREN APPARATUS

suited for a particular type of investigation. A diagram of the model used in this investigation is shown in Fig. A.

The arrangement of the basic apparatus is shown in part *A* of Fig. B. Light from a high intensity source *L* is focused by a condensing lens on the small mirror, M_1, located at the center of curvature of a large spherical mirror, M_2. The light then goes to the large mirror and is reflected back to the cut-off *S* just slightly offset from M_1. A viewing screen *P* is placed after the cut-off.

If an object with an index of refraction differing from that of air is placed between M_2 and *S*, as shown in part *B* of Fig. B, some light rays will be bent so that they will go past the cut-off and form an image on the screen. Therefore, if streams of heated air are passed through the field of view, an image will appear on the screen and will give a picture similar to smoke streams. The unheated air in the stream will not affect the light so that the field will be dark but the heated air will cause light streaks.

Different methods of heating the air have been used by investigators, depending upon the effect desired. If sparks are used whose interval is known, the image appears as dots in stream lines and the spacing indicates the

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velocity. When hot wires are used, a steady stream image is produced so that actual flow patterns may be studied.

The light source used was a 50 cp. automobile head light bulb made to operate on 6 volts but operated on 10 volts to give more light. The light was focused on a metal diaphragm with a 1/64 hole by an achromatic condensing lens. This produced the effect of having a real light source 1/64 in. diameter located at the focal point of the large mirror. The small mirror was made of optically flat glass surfaced on the front. The large spherical mirror ground on a 12-ft radius was 12 in. in diameter.

The cut-off must be arranged according to the direction of flow because the heated air stream has the effect of bending the light in only one direction. Since, in this investigation, the flow turned through 90 deg the cut-off used was in the shape of an L with the sides making an angle of 90 deg with the two directions of flow in the model.

Das Toeplersche Schlierenverfahren, by Dr.-Ing. Hubert Schardin. (*Farschungsheft* 367, July, August, 1934.)

A Method of Air Flow Cinemotography Capable of Quantitative Analysis, by H. C. H. Townend. (*Journal of the Aeronautical Sciences*, Vol. 3, No. 10, August, 1936.)

Improvements in the Schlieren Method, by H. G. Tayler, J. M. Waldrom. (*Journal of Scientific Instruments*, Vol. 10, 1933.)

Improvements in the Schlieren Method of Photography, by H. C. H. Townend. (*Journal of Scientific Instruments*, Vol. 11, 1934.)

No. 1217

TREND CURVES FOR ESTIMATING PERFORMANCE OF PANEL HEATING SYSTEMS

By B. F. Raber * and F. W. Hutchinson,** Berkeley, Calif.

THE performance of a panel heating system can be determined with reasonable accuracy by carrying out a detailed analysis based on rational principles.[1] Likewise, a correlation of experimental data from other installations would be expected to afford a background from which one might estimate the performance of a new system. Unfortunately, however, each of these methods is subject to severe objections: The rational analysis is difficult, laborious and time-consuming while the experimental correlation is limited through lack of satisfactory data from a great enough number of different installations and is further handicapped by the technical problem of extrapolating such data to the conditions of a new design.

Basically, the increased complexity of calculations for the panel system over similar calculations for other types of heating systems arises almost entirely from the fact that, with panel heating, selection of a fixed design inside air temperature is not admissible nor possible. As the MRT (*mean radiant temperature*) increases, the corresponding inside air temperature for a constant sensation of warmth must decrease. The MRT is influenced by the following factors: panel surface temperature, outside air temperature, location of heating surface, ventilation rate and heat transmission characteristics of the structure. Thus any change in one or more of the factors outlined will result in a change of MRT and a consequent variation in the value of the inside air temperature corresponding to optimum comfort.

The essential point of difference between panel heating and other methods of heating is the reduction of the inside air temperature which must occur (for comfort) when the fraction of energy transferred by radiation is increased. As the radiant fraction becomes greater the MRT of the enclosure must increase and the inside air temperature decrease; as the convective fraction increases the opposite sequence must occur. Such considerations suggested that an acceptable criterion of effectiveness for radiant heating systems should

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[1] Panel Heating and Cooling Analysis by R. F. Raber and F. W. Hutchinson. (ASHVE Transactions, Vol. 47, 1941.)

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, St. Paul, Minn., June, 1942.

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indicate the difference between the MRT and the inside air temperature. Such a criterion is defined by the equation,

$$D = \text{MRT} - t_a \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where D is the depression of inside air temperature, t_a, below the mean radiant temperature.

A knowledge of the magnitude and manner of variation of D is a first requisite in the design or performance analysis of a proposed or existing panel heating system. If the depression is large, the heating load on the panel heated structure may differ greatly from the equivalent heating load for a convective system; conversely, if D is small the load will be approximately the same for either panel or convective heating and the need for precise panel analysis will be obviated.

With respect to operating performance, D is significant for two reasons:

1. Systems for which D is small throughout the operating range can be thermostatically controlled at constant air temperature in exactly the same way as convective heating systems. When D varies widely with operating conditions, the optimum inside air temperature must likewise vary, and simple thermostatic control based on constant air temperature cannot be used effectively. In the past, ill-advised criticism of radiant heating has resulted from systems having large variation in D, yet controlled to operate at a constant air temperature. In such cases room conditions inevitably will be too warm for all operating loads greater than the one corresponding to the thermostat setting and too cool for all operating loads less than that corresponding to the particular condition for which the selected air temperature does correspond to optimum comfort.
2. Two of the most important advantages claimed for radiant heating are the greater feeling of freshness and the reduced ventilation heating load which accompany operation at reduced air temperature. Thus changes in D are indicative of the varying effectiveness of panels, under different design or operating conditions, in realizing the advantages which are primarily responsible for their use. Any change in design or application which serves to reduce D likewise serves to reduce the effectiveness of the panels in transferring a maximum fraction of energy by radiation, and hence causes the panel system to approach more closely the performance of a conventional convective system. Hence, D varies directly (but not proportionately) with panel effectiveness; the greater D becomes, the more closely does the panel system approach the performance of an ideal radiant heating system.

TREND CURVES

Based on D as an acceptable criterion of panel performance, the intent of this paper is to present a series of simplified general trend curves to assist the designer in estimating the effect on performance of changes in any one or more of the five principal design factors. Numerically, the magnitude of D and its rate of change with any one variable depend on the specific conditions of the particular structure for which the design or analysis is intended. For estimating purposes, however, numerical values are frequently unnecessary, but information is needed as to the general effect on performance of a proposed change in some one factor.

The trend curves here presented are numerically significant only for the particular structure used in the calculations, but the slope of each curve and the per cent change in D with changes in each of the other factors can be taken as representative for any type of structure when operating at any point within the limits of variation which are used in these curves. Judicious

examination of the rate of change of D with each variable makes possible the use of the curves as a rough basis for estimating the numerical magnitude of D for any other structure.

Specifically, the trend curves have been constructed to provide direct and universally valid answers to five questions that recur constantly during the design or analysis of any panel heating system. These are:

1. For a given panel installation, in what way does operating effectiveness vary with the heating load (or, specifically, with the outside air temperature, t_o)?
2. For a given system operating at design outside temperature, how does effectiveness vary with the ventilation rate?
3. For a given structure with fixed ventilation rate and constant outside temperature, how does effectiveness vary with design panel surface temperature (or, conversely, how does effectiveness vary with design panel area)?
4. For fixed ventilation rate, constant outside temperature, and fixed panel area, how does effectiveness vary with the over-all coefficient of heat transmission of the structure?
5. For a given structure with fixed ventilation rate, constant panel area and fixed outside temperature, how does effectiveness vary with the location of the panel (as floor, wall, or ceiling)?

Answers to these five questions are presented in Figs. 1 to 5, respectively, each of which will be discussed in a later section.

CONSTRUCTION OF CURVES

The calculations leading to the trend curves are based entirely on equations taken from an earlier contribution of the authors.[2] For this reason derivations are not included here, but are briefly referred to in the appendix. Need exists, however, for an explicit statement of the values of certain basic constants and fundamental empirical relationships which have been used in the calculations. These are as follows:

1. Film coefficients of heat transfer by *convection only* (Based largely on ASHVE Report No. 1098).

a. For ceiling panels = 0.40 Btu per hr per sq ft per deg F.
b. For wall panels = 0.70 Btu per hr per sq ft per deg F.
c. For floor panels = 1.10 Btu per hr per sq ft per deg F.
d. Equivalent value for all unheated room surfaces = 0.80 Btu per hr per sq ft per deg F.

2. Equivalent film coefficient for transfer by *radiation only:*[3]
= 1.00 Btu per hr per sq ft per deg F.

3. Comfort is considered as being realized in a room with both air and walls of 72 F. The comfort curve for radiant heating is assumed to pass through the point $t_a = 72$ when MRT = 72 and to have slope of unity. Any error which may exist because of this assumption would result in a change of the position, but not the trend of the curves for D.
4. The design outside temperature has been taken as $t_o = 0$ F.
5. The design panel surface temperature has been taken as $t_p = 120$ F.
6. The design ventilation rate corresponds to an air flow of 2.5 cu ft per hour per square foot of inside surface. This is equivalent to one air change per hour in a room 20 ft x 20 ft x 10 ft.
7. The design over-all coefficient of heat transfer is assumed constant for all unheated surfaces, and is taken as numerically equal to 0.250 Btu per hr per sq ft per deg F. This is equivalent to a conductance from the inside surface to the outside air of 0.29 Btu per hr per sq ft per deg F. To simplify the calculations by eliminating the necessity of considering the shape factors of one wall with respect to others, the

[2] Loc. Cit., Note 1.
[3] Loc. Cit., Note 1.

inside surface temperature has been taken as uniform over all unheated surfaces. Calculation from the precise equations indicates that each of these assumptions leads to a conservative value of D. As a first approximation, therefore, one can neglect shape factors and treat any real structure as though it were constructed uniformly of a material having a conductance (calculated from inside surface to outside air) equal to the average weighted conductance of the real structure. In such cases the conductance of inside partitions or floors having no transmission losses would obviously be taken as 0.

The assumptions of relationship 7 obviate the need for considering the effect of shape factors in applying the heat balance equations for performance analysis, but it is important to note that nothing is said as to the uniformity

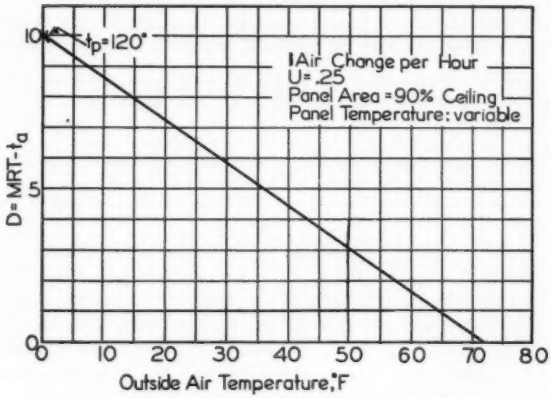

FIG. 1. PANEL EFFECTIVENESS *vs.* OUTSIDE TEMPERATURE

of heating effect *at the occupant*. This paper deals only with the over-all sensation of warmth of an occupant in the room; the comfort of that occupant will also depend on the uniformity of irradiation received at the body surface. This, in turn, depends on proper distribution of panel surface as dictated by analysis from consideration of shape factors of the panels with respect to the occupant.

EFFECT OF LOAD ON PERFORMANCE

The variation of D with load is shown graphically in Fig. 1. Having selected an over-all coefficient of 0.250 and a ventilation rate of 1 air change per hour the necessary area of panel (at 120 F surface temperature) was calculated to provide optimum comfort when the outside temperature is 0 F. A panel area equal to 22.5 per cent of the total inside surface (90 per cent of the ceiling area in a room 20 ft x 20 ft x 10 ft) was found to be required and it was then assumed that this entire area would remain in operation at all higher outside temperatures. With constant panel area the panel surface temperature must obviously vary from a maximum of 120 F at 0 F outside,

to 72 F at 72 F outside. Analysis showed the variation of D with t_o to be a straight line with maximum $D = 10$ F when greatest load occurs and D reaching 0 as the heating load approaches 0.

The curve of Fig. 1 is significant for the following reasons:

1. Since D increases with load, the effectiveness of any panel system in permitting operation with reduced inside air temperature is greatest when the outside temperature is a minimum. This means that the saving on ventilation air heating becomes progressively more significant as the average load increases. Thus the advantage of reduced ventilation load as an argument for panel heating takes on greater significance in localities where the heating season is extended and the load severe than in climates where operation is usually at low load.

2. From the standpoint of health and comfort the desirability of reduced inside air temperature increases as the outside temperature drops. As shown in Fig. 1, a panel system provides the greatest reduction at the time when that reduction is most welcome. With very low outside temperatures, reduced inside air temperatures (with optimum comfort) alleviate to some extent the low humidity condition found in heated rooms, reduce the feeling of stuffiness associated with entrance into a heated space from the outside, and possibly serve to reduce the shock effects associated both with entrance into and exit from a heated room. This latter point is subject to debate, since the heat balance at the body surface is the same regardless of air temperature, but, beyond doubt, the change in conditions in the mucous membranes of the nose and throat which are acted on by the room air, but not affected by the MRT, would be less severe in a panel heated room.

3. An additional factor of interest evident from this Fig. 1 is that tests of panel installations should always be conducted when the outside temperature is low. One determination of D at a low outside temperature will permit drawing accurately the D *vs.* t_o line for the entire operating range, but if the test is conducted at a high outside temperature, the value of D will be small, the numerical value of the experimental error unchanged, and the resultant error from extrapolating the D curve to lower outside temperatures may be very great. Because the variation of D with load is so large, test results from different installations cannot be compared correctly unless the data are first adjusted for differences in outside temperature at the time of the tests.

4. The conditions for which Fig. 1 was constructed are reasonably representative of average practice. Since the value of D at design outside conditions is 10 F, evidently the analysis of the panel system for such a structure based on principles of standard convective heating design might lead to serious error. In the case of this particular structure, however, the convective analysis would be conservative since calculation shows that the rational analysis permits a 5 per cent reduction in panel area over that which would be required if calculations were based on usual convective heating load determinations.

EFFECT OF VENTILATION RATE ON PERFORMANCE

Fig. 2 brings out the very significant fact that the ventilation rate is the most important single factor in determining the performance of a panel heating system. In a room with no mechanical ventilation and 0 infiltration there can be no difference between the MRT and inside air temperature. Obviously, therefore, data from tests on panel heating installations are valueless unless information is presented concerning the ventilation rate at the time of the tests. Regardless of construction, load, outside temperature or panel area, the value of D must approach 0 as the ventilation rate approaches 0. The only exception to this fact is that changes in the film coefficient for convection, with respect to the position of the surface, do influence the heat balance somewhat, and this effect would result in a small D, even at 0 air change. However, this factor is hardly of practical significance and does not alter the

general conclusion that ventilation rate is the most important single factor which influences panel heating effectiveness.

The validity of the conclusion that D must be 0 (neglecting effects of changing film coefficients) when there is no ventilation, can be visualized readily in terms of equilibrium conditions in a sealed room. In such an enclosure heat would flow by convection from the panel to the air and from

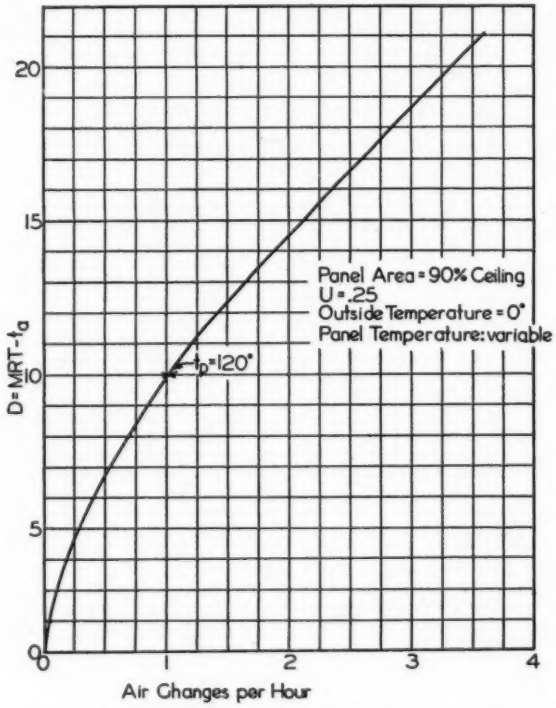

FIG. 2. PANEL EFFECTIVENESS *vs.* VENTILATION RATE

the air to the unheated surfaces; the air temperature would reach an equilibrium value at some point between panel and wall surface temperatures. Suppose that this equilibrium air temperature were less than the MRT. By the concept of MRT as an average or equivalent uniform surface temperature, one could set up an equivalent room with all surfaces at a temperature equal to MRT, but if t_a were less than MRT there would be a flow of heat from the walls to the inside air which then could not be in equilibrium; it follows, therefore, that there cannot be a temperature difference between MRT and t_a.

The conditions for which Fig. 2 was plotted are similar to those for Fig. 1 and the D, 10 F, corresponding to 1 air change per hour, checks Fig. 1.

Since the panel area is assumed constant, the panel surface temperature must vary as a function of ventilation rate; for more than 1 air change per hour the panel temperature will exceed 120 F while for lower ventilation rates the temperature will be less than 120 F.

The increase of D with ventilation rate is seen to be very rapid, the D corresponding to 3½ air changes per hour (specifically, 8.75 cu ft of ventilation air per hour per square foot of inside surface in a room 20 ft x 20 ft x 10 ft) being in excess of 20 F and the air temperature for optimum comfort under such conditions being therefore less than 62 F. The value of a large D as a means of reducing the ventilation air heating load is obviously most important

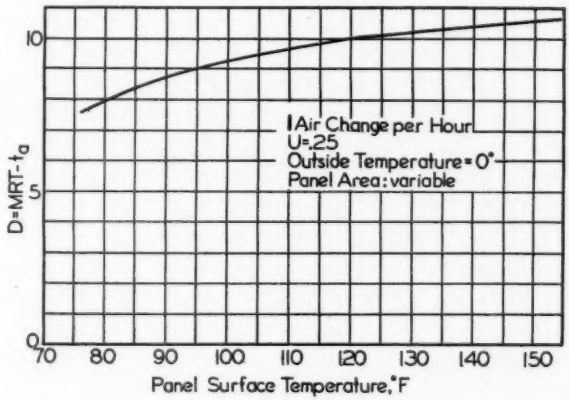

FIG. 3. PANEL EFFECTIVENESS *vs.* SURFACE TEMPERATURE

in structures where the number of air changes is high; the figure also makes evident that the per cent saving on ventilation load through use of panel heating increases as the ventilation load becomes greater. However, even at ventilation rates corresponding to residential infiltration (½ to 1½ air changes per hour) the magnitude of D remains great enough to be of importance in designing and controlling a panel heating system.

EFFECT OF DESIGN PANEL SURFACE TEMPERATURE ON PERFORMANCE

The question has frequently arisen as to whether low temperature panels of large area are more effective than small area, high temperature panels. Fig. 3 was constructed to show the effect on D of varying the design panel surface temperature when all other factors remain the same. Calculations for this figure were based on a constant value of the film transfer coefficient by convection from the panel surface; this condition was considered justified in that consideration of the variation of this factor would have slightly *reduced* the slope of the D *vs.* t_p curve and therefore have indicated an even smaller importance of panel temperature than is now evident from Fig. 3.

Referring to this figure, panel surface temperatures were investigated from 77 F up to 205 F; the value of D at 205 F was found to be less than 11 F, indicating that beyond 150 F panel temperature has practically no effect on the air temperature depression. Even in the range from 90 F to 160 F, corresponding to the extreme limits of present practice, the change in D is too small to be of real significance. The conclusion is unavoidable that heated

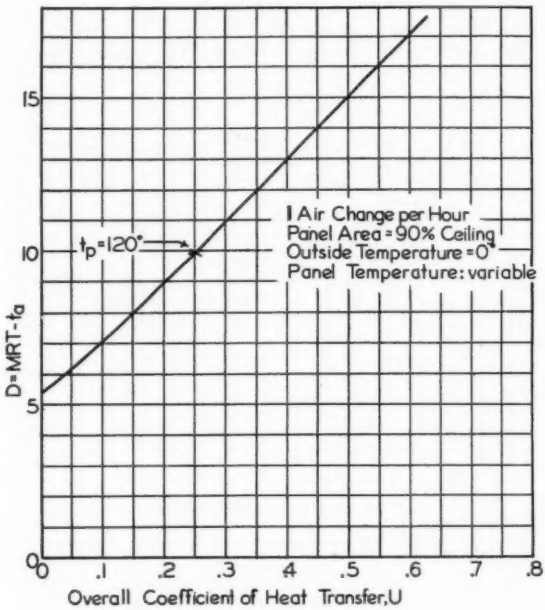

FIG. 4. PANEL EFFECTIVENESS *vs.* TYPE OF BUILDING CONSTRUCTION

surface temperature has very little effect on the operating performance of a panel system.

In connection with the temperature range of Fig. 3 one fact requires comment. The area of a panel operating at 80 or 90 F would exceed the ceiling area of the 20 ft x 20 ft x 10 ft room. This would mean that part of such a panel surface would operate as a wall rather than a ceiling unit. To permit direct comparison of data from all five figures this discontinuity effect was ignored and it was assumed that the installation would be in a room of such shape that the requisite panel area could be installed entirely in the ceiling. The fact is important, however, because it indicates existence of a lower limit to panel temperature in a given room (with given design outside conditions) beyond which the designer cannot go without using a split ceiling-wall panel installation.

EFFECT OF BUILDING CONSTRUCTION ON PERFORMANCE

Fig. 4 is based on the same general conditions as Fig. 1, that is, a panel operating at temperature 120 F establishes comfort in a structure for which the uniform over-all coefficient of heat transfer is 0.250 and the ventilation rate is 1 air change per hour. Outside temperature is constant at 0 F and the D for this set of conditions is $= 10.0$ F. Considering the panel area as fixed and allowing the surface temperature to change as a function of the

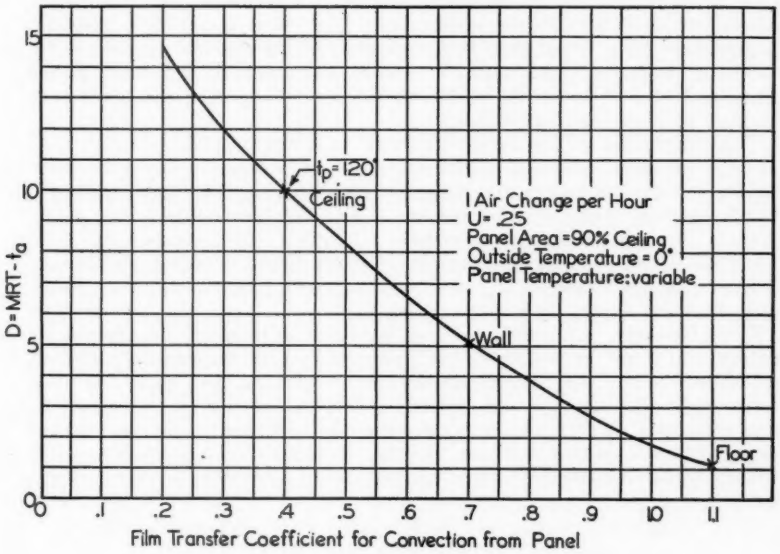

FIG. 5. PANEL EFFECTIVENESS *vs.* PANEL LOCATION

over-all coefficient of heat transfer a relationship is established between D and the type of construction.

As insulation is added D decreases until for the perfectly insulated structure it assumes a minimum value based on 0 transmission losses and a heat balance between the inside surface and the ventilation air passing through the room. Note that the minimum value of D is determined by the ventilation rate, the outside temperature and the panel surface temperature. From previous curves one would expect this minimum D to become greater with increase in panel temperature or ventilation rate and with decrease in outside temperature.

The conclusion is evident that the effectiveness of panel installations is greatest in poorly insulated structures. This view is somewhat at variance with accepted opinion since it suggests that residences may present greater advantages for panel heating than do larger structures. Recent experience

in the United States seems to substantiate this conclusion, but insufficient field data are yet available to permit experimental investigation of the analytical results.

EFFECT OF PANEL LOCATION ON PERFORMANCE

The selection of floor, wall, or ceiling panels for a given installation is influenced by many economic and construction factors which may outweigh in importance the consideration of performance. When, however, performance is the controlling feature there is every reason to believe that ceiling panels are, by far, preferable to any other type. Fig. 5 considers the same structure as in Fig. 1 and plots D as a function of the convection film transfer coefficient from the panel surface. Points are noted on the curve corresponding to the values of the coefficient which would exist if the panel were installed in floor, wall and ceiling.

The construction of the curve was based on constant panel area with consequent reduction in surface temperature as the convection increases. For the three points marked on the curve the respective surface temperatures are 120 F (ceiling), 112 F (wall) and 104 F (floor). Referring to Fig. 3, the drop in D corresponding to a reduction in ceiling panel temperature from 120 F to 104 F would be less than 1 F. From Fig. 5, however, the drop is 9 F so that the influence of panel location is obviously very much greater than the effect of the temperature change.

If performance of a panel system is measured in terms of the fraction of energy transferred by radiation, that is, in terms of the magnitude of D, then it follows directly that ceiling panels come closest and floor panels farthest from the ideal radiant heating system. A well designed floor panel installation may equal in performance a convection heating system and may possess other advantages which justify its use, but it does not in any marked degree possess the characteristics which are said to be desirable in radiant panel heating.

CONCLUSIONS

The trend curves presented in this paper were established for a particular structure by application of the rational equations developed in a previous paper. While numerically significant only for the conditions used in their construction, the curves give certain general qualitative results which can be taken as applicable to any problem involving design or analysis of a panel heating system.

Taking the depression D, of air temperature below mean radiant temperature as a criterion of effectiveness in radiant heating, five conclusions follow from the curves:

1. Greatest advantage is realized from panel heating in localities where the winter outside temperature is low; effectiveness increases with heating load.
2. Ventilation rate is the most important single factor influencing panel performance. At zero ventilation rate there can be no difference between mean radiant temperature and air temperature; at a rate of ½ air change per hour D becomes significant and at higher ventilation rates it assumes large values. The rational expression for ventilation in panel heating is cubic feet of air per unit time per unit inside surface area.

3. The temperature of a panel has but small influence on its performance; in the range from 90 to 150 F increased surface temperatures give a slightly greater D.
4. Effectiveness increases as insulation decreases. This does not mean that insulation is undesirable, but simply that the advantage of panels is greater in poorly constructed buildings.
5. Effectiveness is greatly reduced if the panel is placed in the wall or floor rather than the ceiling. Floor panels have such a small D that their performance can best be estimated by considering them as a form of convective rather than radiant heating system.

Summarizing the foregoing conclusions in terms of the variables which appear in the rational equations: The ceiling is the most effective location for radiant heating panels and the performance of such panels becomes more effective as outside temperatures decrease and as panel surface temperature, ventilation rate, and structure conductance increase.

APPENDIX

The equations used in establishing the trend curves presented in this paper are simplified forms of the basic rational equations derived in an earlier paper.[4] Except where otherwise noted, the nomenclature for the following equations corresponds to that used in the original publication.

Equation I—Heat balance on unheated wall surface: Equation (48) of the original paper established a heat balance on the unheated section of wall surface. This equation was,

$$t_w = \frac{h_R(F_{fw}t_f + F_{iw}t_i + F_{pw}t_p + F_{cw}t_c + F_{gw}t_g) + h_w t_a + C_w t_o}{h_R(F_{fw} + F_{iw} + F_{pw} + F_{cw} + F_{gw}) + h_w + C_w} \quad . \quad . \quad . \quad . \quad . \quad . \quad (48)$$

For the room considered in construction of the trend curves all inside surface is either heated or is unheated surface having constant conductance. The panel area is considered small with respect to the size of the enclosure so that $F_{wp} = 1.0$ and, by the reciprocity theorem, $F_{pw} = A_p/A_w$. Letting $h_R = 1.0$ and $h_w = 0.80$ Equation (48) then becomes,

$$t_w = \frac{A_p t_p + 0.8A_w t_a + C_w A_w t_o}{A_p + 0.8A_w + C_w A_w} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (a)$$

Letting $A_p/(A_p + A_w) = u$ and $A_w/(A_p + A_w) = v$ where u and v are then the fractions of total inside surface which are heated and unheated respectively Equation (a) becomes,

$$t_w = \frac{ut_p + 0.8vt_a + C_w v t_o}{u + 0.8v + C_w v} \quad . \quad . \quad . \quad . \quad . \quad (\textit{Equation I})$$

Equation II—Heat balance on air passing through room: Equation (55) of the original paper establishes a heat balance on the ventilation air passing through the room,

$$t_a = \frac{h_p t_p A_p + h_c t_c A_c + h_g t_g A_g + h_w t_w A_w + h_f t_f A_f + h_i t_i A_i + 0.24Wt_o}{h_p A_p + h_c A_c + h_g A_g + h_w A_w + h_f A_f + h_i A_i + 0.24W} \quad . \quad . \quad . \quad (55)$$

[4] Loc. Cit., Note 1.

Simplifying this equation by the same substitutions used in *Equation I* with the added substitution of,

$$V_c = \frac{13.5W}{A_p + A_w}$$

where V_c is then the cubic feet of ventilation air passing each square foot of inside surface per hour,

$$t_a = \frac{0.4ut_p + 0.8vt_w + 0.0178V_ct_o}{0.4ut_p + 0.8t_w + 0.0178V_c} \quad . \quad . \quad . \quad \textit{(Equation II)}$$

Equation III—Comfort Equation: Fig. 1 of the original paper presented a comfort line defined by the equation,

$$t_a = \phi\,(\mathrm{MRT})$$

Based on the assumptions discussed in an earlier section of this paper the comfort line is considered straight, of unit slope and passing through the point 72 F, 72 F,

$$t_a = 144 - \mathrm{MRT}$$

but,

$$\mathrm{MRT} = ut_p + vt_w$$

$$\therefore t_a = 144 - ut_p - vt_w \quad . \quad . \quad . \quad . \quad . \quad \textit{(Equation III)}$$

Analysis

Equations I, II, and III independently relate the seven variables: t_p, t_w, t_a, t_o, C_w, V_c and u. From three simultaneous equations it is possible to eliminate two variables, as t_a and t_w, to give an equation of the form,

$$u = \phi\,(t_p, t_o, C_w, V_c). \quad . \quad . \quad . \quad . \quad . \quad \textit{(Equation IV)}$$

Fig. 1: Using *Equation IV* with $t_p = 120$ F; $t_o = 0$ F, $C_w = 0.29$ and $V_c = 2.5$ (corresponding to one air change per hour in a room 20 ft x 20 ft x 10 ft) a solution for the panel area becomes possible, yielding $u = 0.225$ (90 per cent of ceiling area in 20 ft x 20 ft x 10 ft room). Then substituting known values of u, V_c, and C_w in *Equation IV* to obtain,

$$t_p = \phi\,(t_o) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \textit{(Equation V)}$$

Equations I, II, and III can be solved to obtain an equation of the form, $D = \phi\ (u, C_w, V_c, t_p, t_o)$ where all variables except t_p and t_o are constants and these two terms are related by *Equation V*. Substituting,

$$D = \phi(t_p)$$

which is the relationship shown in Fig. 1 of the paper.

By a similar method of analysis other factors can be eliminated and the equations established to permit plotting Figs. 2, 3, 4 and 5.

All points on all curves satisfy the three basic panel analysis equations and therefore represent state points corresponding to comfort (as defined in *Equation III*) as well as equilibrium in terms of the heat transfer characteristics of the system.

No. 1218

OVERLOADING OF VISCOUS AIR FILTERS DURING ACCELERATED TESTS

By Frank B. Rowley * and Richard C. Jordan,** Minneapolis, Minn.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the Engineering Experiment Station, University of Minnesota

AIR filters of the unit replaceable type may be classified according to their principle of operation as either viscous or dry. The viscous type of filter usually consists of a fibrous or cellular material coated with an adhesive. The filter pack is usually graded as to density, fiber size, adhesive coating, or some combination of these three for the purpose of progressively cleaning the air as it passes through the filter and for distributing the load throughout the filter. Dust is eliminated from the air by subdividing the air stream in its passage through the filter to increase the possibility that all of the dust particles may come into contact with the adhesive material. The air-dust mixture is made to change directions frequently in its passage through the filter, and the dust is thus centrifuged against the oiled surface.

The dry type of filter works upon a somewhat different principle from that of the viscous type. It generally consists of a finely woven or packed material having comparatively small interstices. In some dry filters the air passages are sufficiently fine that the dust is removed by the screening action. However, it appears that in most cases the major part of the dust is removed as in the viscous filters by impingement and centrifuging.

In the laboratory rating of air filters by the weight or crucible method several difficulties are introduced through the necessity that the tests be accelerated. Since it would be impractical to feed an artificial test dust to a filter in the laboratory at the same rate that it would be subjected to dust in an actual installation, the test filters are subjected to rates of feed 40 to 160 times normal in order that the test may be completed in a reasonable period of time. This requirement of a practical laboratory test apparently has little effect upon the performance characteristics of a dry type of filter as the efficiency and dust holding capacity vary only slightly from low to high rates

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of dust feed. With the viscous type of filter, however, it is possible to feed the dust to the filter faster than the adhesive can absorb the impinged dust particles. The limiting rate of feed during an accelerated test is apparently dependent not only upon the nature of the adhesive but also upon the design of the filter and the physical properties of the dust being used for test purposes. Thus a rate of dust feed which will cause overloading with one type of filter and one type of dust may be satisfactory for the testing of other filters and with other test dusts. The results of overloading are to reduce the average efficiency of the filter and usually, but not always, to increase

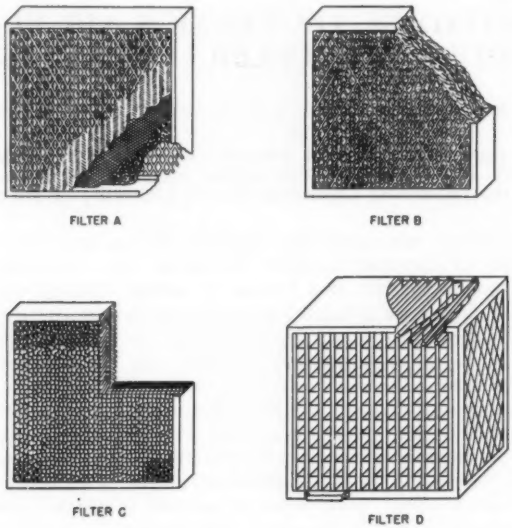

FIG 1. SECTIONAL VIEWS OF AIR FILTERS TESTED

the life and dust-holding capacity. For these reasons there has been a need for extended studies covering the causes of filter overloading and means for its prevention. The present investigations cover the following points:

1. The test conditions under which filter overloading may occur.
2. The physical properties of test dusts which cause filter overloading.
3. Dust components which are readily reproducible and which will not result in fictitious performance characteristics through overloading.

TEST DUST MIXTURES

One of the major problems in the formulation of a standard air filter code for the rating of filters by the weight method has been the determination of a satisfactory artificial test dust. Such a dust must not only simulate air-

borne dust in all of the properties which may affect filter performance, but it must also be reproducible as to its physical properties, must not cause filter overloading at practical rates of dust feed, and must be practical to obtain in sufficient quantities for test purposes.

Various mixtures of synthetic components have been tried at different times and proposed as standard dust mixtures. The original ASHVE Air Filter Test Code specified a mixture consisting of 50 per cent by weight of powdered lampblack, containing a minimum of 97.5 per cent free carbon, and having a minimum bulking value of 3.5 lb per cubic foot, mixed wth 50 per cent by

FIG. 2. VIEW OF AIR FILTER TEST APPARATUS SHOWING AIR TEMPERATURE CONTROL CHAMBER

weight of Pocahontas bituminous coal ash screened to pass a 200 mesh sieve. However, it was shown later [1] that many different carbons having widely varying effects upon air filter performance characteristics would satisfy this requirement. Furthermore, filters of the same type, when tested with different lots of lampblack of the same brand and manufacture, showed efficiency values ranging from 28.1 to 82.4 per cent. It was, therefore, obviously impractical to use a material with such a wide range of test results as a basis for a standard test dust. Such a dust was satisfactory only for comparative performance tests of different filters in which a sufficiently large supply of the mixture was prepared for use throughout the entire series of investigations.

A test dust consisting of 50 per cent Pocahontas ash, 20 per cent Illinois fly ash, 20 per cent lampblack, and 10 per cent Fuller's Earth was selected as a basis for a series of railroad air filter tests made at the University of

[1] ASHVE RESEARCH REPORT No. 1122—Air Filter Performance as Affected by Low Rate of Dust Feed, Various Types of Carbon, and Dust Particle Size and Density, by Frank B. Rowley and Richard C. Jordan. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

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Minnesota in 1937. This dust was the result of a conference held by various manufacturers and users of air filters. This air filter test dust too was satisfactory only for comparative tests as no rigid specifications were set upon the physical properties of the components, and therefore, there was no assur-

TABLE 1—COMPARISON OF FILTER PERFORMANCE CHARACTERISTICS FOR OVERLOAD TESTS

FILTER	PER CENT	DUST MIXTURE	RATE OF FEED, GRAMS/HR	FILTER PERFORMANCE BASED ON 0.4 IN. H_2O FINAL RESISTANCE			FILTER PERFORMANCE BASED ON 1.0 IN. H_2O FINAL RESISTANCE		
				Efficiency, Per Cent	Life, Hours	Dust Holding Cap., Grams	Efficiency, Per Cent	Life, Hours	Dust Holding Cap., Grams
B-I	50 20 20 10	Pocahontas Ash Illinois Fly Ash Lampblack Fuller's Earth	20	81.1	11.5	186.6	84.0	19.9	334.4
B-I	50 20 20 10	Pocahontas Ash Illinois Fly Ash Lampblack Fuller's Earth	40	82.6	5.2	171.9	85.3	9.0	307.2
B-I	80 20	Pocahontas Ash Carbon Dust [a]	40	86.9	7.0	243.0	58.5 [b]	...	...
B-I	80 20	Pocahontas Ash Lampblack	40	83.8	4.2	140.7	86.3	8.8	303.8
C	50 20 20 10	Pocahontas Ash Illinois Fly Ash Lampblack Fuller's Earth	40	72.6	10.2	296.3	70.6	18.4	519.4
C	80 20	Pocahontas Ash Carbon Dust [a]	40	47.4	21.5	408.0	41.5	44.5	738.9
C	80 20	Pocahontas Ash Carbon Dust [a]	20	70.2	16.0	224.6	66.9	32.6	436.2
D	50 20 20 10	Pocahontas Ash Illinois Fly Ash Lampblack Fuller's Earth	40	94.7	7.2	272.8	95.5	9.1	347.6
D	80 20	Pocahontas Ash Carbon Dust [a]	40	96.0	8.6	330.4	96.9	11.8	457.2

[a] Double Bolted Carbon Dust (Binney and Smith Co.)
[b] Average for 18-hour test period.

ance that the dust would be reproducible at a later date or in some other section of the country. A great many other dust mixtures have been proposed at various times, but without exception the stipulations placed upon the physical properties of the components have been inadequate to insure their satisfactory use as a standard test dust.

In 1939 a series of tests was conducted at the Engineering Experiment Station of the University of Minnesota in cooperation with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in order to investigate various materials which might be synthesized and used satisfactorily as components of an artificial test dust. During these tests 15 different types of materials were investigated [2] and as a result it was tentatively recommended that one dust mixture which would be reproducible and, in the main, satisfy the general requirements of a standard dust would consist of a mixture of 80 per cent by weight of Pocahontas ash screened through a 200 mesh sieve and having a jolted density between 0.65 and 0.85 grams per cubic centimeter and 20 per cent by weight of Double Bolted Carbon Dust (Binney and Smith Co.) screened through a 100 mesh sieve and having a jolted density between 0.145 and 0.180 grams per cubic centimeter. This dust was used for some time at the University of Minnesota and was found in general to satisfy the majority of the requirements of a standard dust including that of reproducibility. However, it was noted that under certain conditions of test and with certain specific filters the test results indicated that the filter was being overloaded, and therefore the test results were, to a certain extent, falsified.

TEST PROCEDURE

In order to carry out these investigations on the overloading of viscous air filters, four typical filters as shown in Fig. 1 were chosen for experimental laboratory tests. These may be described briefly as follows:

Filter A—A permanent type of cleanable oil filter, 2 in. thick, with layers of expanded metal and wire screen graded from coarse mesh at the entrance to fine mesh at leaving side.

Filter B—A viscous coated throw-away filter 2 in. thick. The fibrous media were graded in fiber size, density, and oiling from entering to leaving side. A modification of this filter, which has been designated as *Filter B-1* in these tests, was also used in some of the investigations and incorporated curly instead of straight fiber media.

Filter C—A cellular type filter 2 in. thick built in two sections, each with the axis of cells set at 45 deg to the center-line of duct and at 90 deg to each other. The cells on the entering side were of larger dimensions than those on the leaving side of the filter.

Filter D—A filter of cotton media of coarse material on the entering side and glazed on the leaving side. The filter media were accordion plaited in frame to give an area of approximately 12 times the cross-sectional area of the air stream.

The weight, or crucible, method of testing and rating air filters which has been described in detail in a previous publication [3] was used in these investigations. Throughout the investigations the filter face air velocity was maintained at 300 fpm, and the dust was fed to the filter at rates of 20 or 40 grams per hour, dependent upon the specific tests. In the majority of the investigations determinations were made of the efficiency, life, and dust-holding capacity, with the latter two characteristics based upon a limiting resistance of either 0.4 in. or 0.1 in. of water, or a resistance rise of 0.15 in. of water. In many cases both limiting resistances were used to define filter life and

[2] ASHVE RESEARCH REPORT No. 1143—A Standard Air Filter Test Dust, by Frank B. Rowley and Richard C. Jordan. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

[3] Air Filter Performance as Affected by Kind of Dust, Rate of Dust Feed, and Air Velocity through Filter, by F. B. Rowley and R. C. Jordan. (ASHVE TRANSACTIONS, Vol. 44, 1938.)

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dust holding capacity on the same tests in order to indicate the degree of filter overloading at different dust loads.

As evidence of filter overloading during an accelerated test three different criteria were used:

1. If, during a filter test, there was a marked drop in efficiency as the test proceeded, the filter was considered to have been overloaded.
2. If a filter was tested for a period of hours and then allowed to stand without test during a second period and if, upon resuming the filter tests, there was a marked increase in efficiency and a decrease in resistance, the filter was considered to have been overloaded.
3. If two identical filters were tested at different rates of dust feed and if, after correcting for the rates of feed, there was a marked difference in the performance

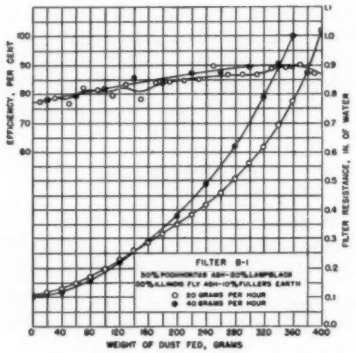
EFFICIENCY, PER CENT
FILTER RESISTANCE, IN. OF WATER
WEIGHT OF DUST FED, GRAMS
FILTER B-1
50% POWDERED ASH—50% LAMPBLACK
50% ILLINOIS FLY ASH—50% FULLER'S EARTH
O 20 GRAMS PER HOUR
● 40 GRAMS PER HOUR

FIG. 3. PERFORMANCE CURVES FOR *Filter B-1* AT TWO RATES OF DUST FEED

characteristics, then the filter tested at the higher rate of feed was considered to have been overloaded.

All three of these criteria were evidence that the dust was being fed to the filter faster than it could be absorbed by the viscous coating. However, it should be realized that the second criterion noted is evident to a small degree on practically all accelerated filter tests conducted at practical rates of dust feed and that overloading accompanying any such minor indications usually has only a slight effect upon the performance characteristics.

One series of performance tests was run to determine the effect of air temperature upon filter performance and overload. The apparatus used for these tests was identical with that used during the ordinary dust tests, with the exception that an air-mixing chamber, as shown in Fig. 2, was constructed on the inlet of the test duct. This mixing chamber was so designed that air could be drawn directly from the room, from the outside, or through a heater coil. Thus it was possible during cold winter days to run air filter tests at temperatures ranging from 0 F to 130 F. During tests the air temperatures were measured by means of a direct-reading potentiometer and

four copper-constantan thermocouples located in the air stream in front of the filter.

OVERLOAD TESTS

In order to show those conditions under which overloading can be obtained, a series of performance tests was made on *Filters B-1, C,* and *D* with various dust mixtures. *Filter B-1* was tested at rates of feed of 20 and 40 grams per hour, using the 50-20-20-10 dust mixture containing 50 per cent Pocahontas ash, 20 per cent Illinois fly ash, 20 per cent lampblack, and 10 per cent Fuller's Earth. In addition, two tests were made on *Filter B-1* at 40 grams per hour using an 80-20 dust mixture containing 80 per cent Pocahontas ash and 20 per cent Double Bolted Carbon Dust, and an 80-20 mixture containing 80 per

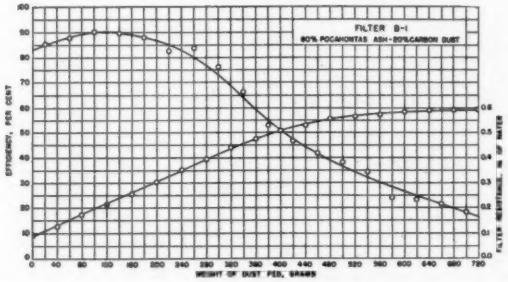

FIG. 4. PERFORMANCE CURVES RESULTING FROM OVERLOADING OF *Filter B-1.* DUST FEED, 40 GRAMS PER HOUR

cent Pocahontas ash and 20 per cent lampblack. *Filter C* was tested at 40 grams per hour and 20 grams per hour using the 80 per cent Pocahontas ash and 20 per cent Double Bolted Carbon Dust mixture and also at 40 grams per hour using the 50-20-20-10 dust mixture. *Filter D* was tested at 40 grams per hour using the 50-20-20-10 dust mixture, and also with the 80 per cent Pocahontas ash, 20 per cent Double Bolted Carbon Dust mixture. All of these tests were made at a filter air face velocity of 300 fpm and to a final limiting resistance of 1.0 in. of water. The results of these tests are recorded in Table 1, and in the performance curves shown graphically in Figs. 3, 4, 5, 6, and 7.

The comparison between the performance characteristics for *Filter B-1,* tested on the 50-20-20-10 dust mixture at rates of feed of 20 and 40 grams per hour, is shown in Fig. 3 and Table 1. Regardless of the high limiting resistance, there is little, if any, indication of overloading at the higher rate of dust feed when using this test dust. The average efficiencies are within 1.5 percentage units when based on either a final limiting resistance of 0.4 in. or 1.0 in. of water. Both the filter life and dust-holding capacity check as closely as can be expected considering the normal variations to be found

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between filters of the same design and manufacture. These tests bear out all previously reported tests [4] which have shown that there is little variation in performance characteristics of the same filter when tested with this dust at rates of feed varying from 20 to 60 grams per hour.

Fig. 4 and Table 1 show the results of the tests made on *Filter B-1* with the 80 per cent Pocahontas ash and 20 per cent carbon dust mixture at a 40 gram per hour rate of feed. These results indicate that the filter became overloaded shortly before it reached a resistance of 0.4 in. of water and that thereafter it was unable to absorb the dust as rapidly as it was fed to the filter. After passing this resistance, the efficiency curve dropped rapidly and the resistance curve levelled off at approximately 0.6 in. of water. It would have been impractical to continue this test any further, and obviously any reported performance characteristics would have been fictitious as the effi-

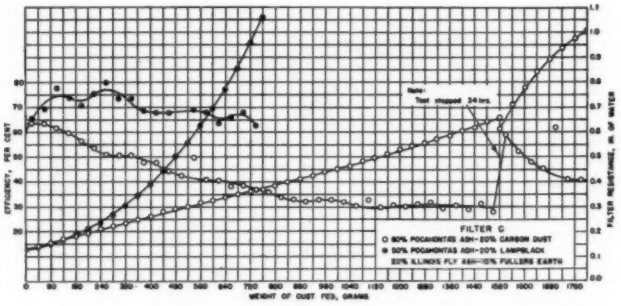
EFFICIENCY, PER CENT
FILTER RESISTANCE, IN. OF WATER
WEIGHT OF DUST FED, GRAMS
FILTER C
80% POCAHONTAS ASH—20% CARBON DUST
80% POCAHONTAS ASH—20% LAMPBLACK
20% ILLINOIS FLY ASH—80% FULLERS EARTH

FIG. 5. PERFORMANCE CURVES FOR *Filter C* ON AN OVERLOADING AND A NON-OVERLOADING DUST MIXTURE. DUST FEED, 40 GRAMS PER HOUR

ciency had dropped from a high of 90 per cent to a low of approximately 18 per cent. The average efficiency, when based upon a limiting resistance of 0.4 in. of water, was 86.9 per cent; when the test was continued to a resistance of 0.6 in. of water, however, the average efficiency dropped to approximately 58.5 per cent. No appreciable overloading took place until the filter pressure drop had exceeded 0.4 in. of water, and if the test had been concluded at that time, the results would probably have been considered satisfactory.

The results of the performance tests made on *Filter B-1*, with the dust mixture consisting of 80 per cent Pocahontas ash and 20 per cent lampblack, at a dust rate of feed of 40 grams per hour, are presented in Table 1. All previous tests had indicated that the dust component causing the overloading was the Double Bolted Carbon Dust, and for this reason lampblack was substituted for carbon dust in the 80-20 dust mixture. The results of this test indicated no overloading, and performance characteristics based on either a

[4] Loc. Cit., Note 3.

0.4 in. of water or 1.0 in. of water final resistance were very close to those determined for the 50-20-20-10 dust mixture on this same filter. Apparently the Pocahontas ash, the Illinois fly ash, and the Fuller's Earth all had about the same effect on the performance characteristics, and therefore the substitution of Pocahontas ash for the Illinois fly ash and the Fuller's Earth had little effect upon the resulting performance characteristics.

The effects of overloading are clearly shown in the results obtained on *Filter C* and presented in Figs. 5 and 6 and in Table 1. The performance characteristics using the 50-20-20-10 dust mixture appeared reasonably normal

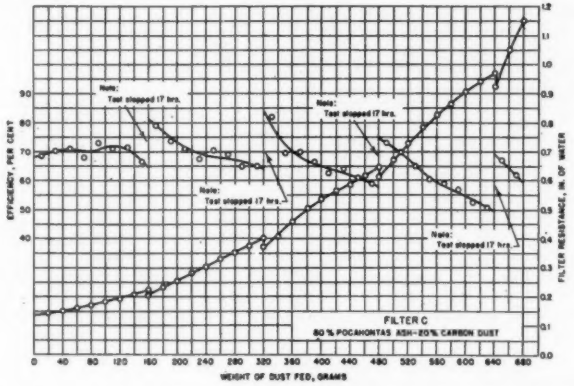
EFFICIENCY, PER CENT
FILTER RESISTANCE, IN. OF WATER
WEIGHT OF DUST FED, GRAMS
Note: Test stopped 17 hrs.
Note: Test stopped 17 hrs.
Note: Test stopped 17 hrs.
Note: Test stopped 17 hrs.
FILTER C
80 % POCAHONTAS ASH-20% CARBON DUST

FIG. 6. PERFORMANCE CURVES RESULTING FROM OVERLOADING OF *Filter C* DURING AN INTERMITTENT TEST. DUST FEED, 20 GRAMS PER HOUR

and showed only a slight overloading as indicated by a small drop in the efficiency curve. However, the same filter, tested at the same rate of dust feed but with the 80 per cent Pocahontas, 20 per cent carbon dust mixture, definitely indicated that overloading was taking place almost as soon as the test had started. The 80-20 dust mixture was tested at two rates of feed, 20 and 40 grams per hour. At the higher rate of feed the test was conducted for a period of 38 hours after which there was a lapse of 34 hours in which no dust was fed to the filter. The test was then resumed and continued until the filter resistance had reached 1.0 in. of water. As indicated in Fig. 5, there was a gradual reduction in efficiency from 64 per cent to 28 per cent during the feeding of 1,520 grams of dust. After the filter had stood for 34 hours without test, the efficiency immediately rose from 28 to 59 per cent upon resuming the test, and then again dropped to 40 per cent. During the 34 hour period during which the test was stopped, the filter resistance dropped from 0.66 in. to 0.61 in. of water. Furthermore, upon resuming the test there was a marked increase in the slope of the resistance curve, giving additional proof of overloading. With both the 50-20-20-10 and the 80-20

dust mixtures fed to *Filter C* at the rate of 40 grams per hour, the initial efficiencies were approximately the same. However, after the resistances had reached 0.4 in. of water, the average efficiency in the case of the 50-20-20-10 mixture was 72.6 per cent, and in the case of the 80-20 mixture was 47.4 per cent. Based upon a final limiting resistance of 1.0 in. of water, the 50-20-20-10 mixture had an average efficiency of 70.6 per cent, and the 80-20 mixture an average efficiency of 41.5 per cent.

Fig. 6 and Table 1 show the results of the test made on *Filter C* with the 80 per cent Pocahontas ash, 20 per cent carbon dust mixture at a 20 gram per hour rate of dust feed to determine whether lowering the rate of feed would eliminate or reduce the overloading. In addition to reducing the rate of feed the test was conducted in 8-hour periods interspersed by periods of

TABLE 2—PROPERTIES OF AIR FILTER ADHESIVES

OIL NO.	FILTER	POUR POINT, DEG F	SAYBOLT VISCOSITY, SEC.		
			70 F	100 F	130 F
1	A	10	4011	1222	486
2	A	− 5	946	320	138
3	B	20	1170	426	230
4	C	60	...	2040 (Furol tube)	805 (Furol tube at 122 F)

rest. Although the result was to increase greatly the average efficiency, based on both the 0.4 in. and the 1.0 in. of water limiting resistances, and to reduce both the life and dust-holding capacity, as compared with the test conducted at 40 grams per hour, there was still much evidence of overloading. During periods of rest there were marked rises in efficiency and drops in resistance, which increased in magnitude as the tests progressed. Furthermore, towards the end of the test, the curvature of the resistance graph changed from positive to negative.

The results of the performance tests on *Filter D* using the 50-20-20-10 and the 80-20 dust mixtures are shown in Fig. 7 and Table 1. Neither of these tests gave any indication of filter overloading, a result which would be expected with dry filtering media and no oil adhesive involved. The efficiency curves were practically identical, although the filter life was somewhat greater with the 80-20 dust mixture. However, variations of this magnitude are often found on supposedly identical filters tested under the same conditions.

The results of these tests indicate that filter overloading, resulting in fictitious performance characteristics, may result when viscous filters are tested under certain conditions. There is apparently no danger of overloading with dry filters, but with those designed upon the adhesive impingement principle overloading may occur if the rate of dust feed is more rapid than the rate at which the adhesive can absorb the dust. Such a condition is apparently brought about if certain types of test dusts are used, if the filter adhesives

are extremely viscous and incapable of absorbing dust rapidly, or in some cases if the final limiting resistance is arbitrarily placed much beyond that to which the filter may normally be subjected under good operating conditions. It is probable that under all accelerated filter tests there is a slight overloading, but any abnormal overloading which would appreciably affect the resulting performance characteristics will be indicated either by a decrease in the efficiency as the test progresses, a sudden change in the efficiency and resistance after an interruption of appreciable duration during a test, or in a

TABLE 3—AIR FILTER PERFORMANCE CHARACTERISTICS AT DIFFERENT AIR TEMPERATURES

(*Dust mixture* 50–20–20–10. *Face Air Velocity* 300 *fpm.* *Dust feed* 40 *grams per hour.*)

AIR TEMP. DEG F	PERFORMANCE CHARACTERISTIC	*Filter A* (OIL NO. 1)	*Filter A* (OIL NO. 2)	*Filter B*	*Filter C*	
30	Efficiency, per cent	74	76	84	62	64[a]
	Life, hours	5.7	6.2	3.6	13.9	11.0[a]
	Dust-Holding Capacity, grams	170	187	122	345	282[a]
70	Efficiency, per cent	77	80	85	72	73
	Life, hours	7.0	8.0	3.2	6.9	6.4
	Dust-Holding Capacity, grams	215	256	109	198	186
130	Efficiency, per cent	78	81	85	76	74
	Life, hours	9.2	8.0	5.0	8.8	8.6
	Dust-Holding Capacity, grams	286	261	168	268	254

[a] 20 grams per hour rate of dust feed. Life based upon 40 grams per hour rate of feed.

change in the performance characteristics of similar filters when tested with the same test dust but under different rates of dust feed.

FILTER TESTS AT DIFFERENT AIR TEMPERATURES

A series of tests was made with *Filters A*, *B*, and *C* to determine the effects of air temperature upon filter performance and overloading. The dust chosen for these tests was the 50-20-20-10 mixture which had previous shown [5] no appreciable overloading on any filters tested at rates of feed varying from 2 to 100 grams per hour. In all tests the face air velocity was maintained at 300 fpm, and, with one or two exceptions as noted, the rate of dust feed was 40 grams per hour. *Filter A*, a permanent, cleanable type of filter was tested with both a light and a heavy oil, some of the properties of which are presented in Table 2 along with those for the adhesives used on the other filters tested.

[5] Loc. Cit., Note 1.

One series of tests was made at air temperatures of 30, 70 and 130 F, in which the average efficiency, the life, and the dust-holding capacity were determined for a resistance rise of 0.15 in. of water. A second series of tests was conducted at air temperatures of 0, 30, 70 and 130 F with determinations made only of the average efficiency for a three-hour test period. It was impossible to obtain complete filter characteristics at the 0 F temperature, as this low temperature could not be maintained over a sufficient number of hours to complete the performance tests.

Table 3 shows the complete performance characteristics determined over a 0.15 in. of water increase in filter resistance. The actual performance curves are not included because of lack of space. With the exception of *Filter C*, these tests indicated no significant differences in the efficiencies at the different air temperatures. *Filter C*, however, when tested at a rate of 40 grams per hour, dropped from an efficiency of 72 per cent at 70 F to an

TABLE 4—AIR FILTER EFFICIENCY AT DIFFERENT AIR TEMPERATURES
(*Dust mixture* 50–20–20–10. *Face Air Velocity* 300 *fpm.* *Dust feed* 40 *grams per hour.*)

AIR TEMP. DEG F	*Filter A* (OIL NO. 1)	*Filter A* (OIL NO. 2)	*Filter B*	*Filter C*	
0	65	75	85		57[a]
30	75	76	84	65	68[a]
70	77	78	85	71	70
130	78	79	84	72	70

[a] 20 grams per hour rate of dust feed.

efficiency of 62 per cent at 30 F. In addition there was a corresponding increase in the dust-holding capacity and life at the lower temperature, and the efficiency curve showed a definite drop in efficiency as the test progressed. The 50-20-20-10 dust had on no previous occasions shown any indications of overloading even at considerably higher rates of feed, but it appeared that at this low temperature the adhesive had become so viscous that it could no longer absorb the dust as rapidly as it was fed. For this reason *Filter C* was again tested with this dust at 30 F, but at the reduced rate of feed of 20 grams per hour, in order to see if overloading could be prevented or reduced. The resulting performance characteristics showed a slightly higher average efficiency and somewhat lower life (corrected for difference in rate of dust feed) and dust-holding capacity than those at the same temperature but at the 40 grams per hour rate of dust feed. However, there was still some overloading at this 20 grams per hour rate of feed as evidenced by the lower efficiency and greater life and dust-holding capacity in comparison with the 70 F test, and by a slight drop in the efficiency curve as the test progressed.

The tests conducted on *Filter A* with oils Nos. 1 and 2 and on *Filter B* showed no indications of overloading at low temperature. Although there were slight drops in the efficiency when lowering from 70 F to 30 F, there were also decreases in life and dust-holding capacity, and such corresponding decreases are not typical of an overloaded condition. In addition, analyses

of the efficiency curves showed no indications of any overloading. It would therefore appear that the differences in the performance characteristics at the different temperatures for these three filters were indicative of the true differences to be found under non-overloaded conditions.

All filters showed equal or slightly higher efficiency and higher life and dust-holding capacity at the 130 F temperature than at the 70 F temperature. This would indicate that with these filter designs somewhat better performance can be expected at the higher temperatures. *Filter B* showed the lowest

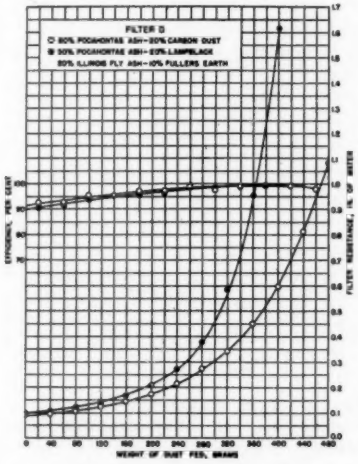
FILTER D
80% POCAHONTAS ASH—20% CARBON DUST
50% POCAHONTAS ASH—50% LAMPBLACK
80% ILLINOIS FLY ASH—10% FULLERS EARTH
EFFICIENCY, PER CENT
FILTER RESISTANCE, IN. OF WATER
WEIGHT OF DUST FED, GRAMS

FIG. 7. PERFORMANCE CURVES FOR *Filter D* ON TWO DUST MIXTURES. DUST FEED, 40 GRAMS PER HOUR

life and dust-holding capacity at a temperature of 70 F, but the differences in magnitude of the performance characteristics between 30 F and 70 F were not sufficiently great to warrant any conclusion.

Table 4 presents the results of the efficiency determinations based on the three-hour tests made at the different air temperatures. Both *Filter A* with oil No. 2 and *Filter B* showed substantially the same efficiencies throughout the entire range of temperatures from 0 F to 130 F; however, *Filters C* and *A* with the more viscous adhesives showed marked reductions in efficiencies at 0 F over those determined at 30 F. The test on *Filter C* was made at a 20 grams per hour rate of dust feed and should therefore be compared only with the test made at 30 F at the same rate of feed. The efficiency of *Filter C* dropped from 68 per cent to 57 per cent and that for *Filter A* with oil No. 1 from 75 per cent to 65 per cent when the temperature was reduced from 30 F to 0 F. As it appeared probable from the first series of air temperature tests that the reductions in efficiency for *Filter C* at 30 F over

the tests at 70 F were the result of overloading, these further reductions in efficiency at 0 F were probably also caused by overloading. However, there is no direct evidence to substantiate this, as it was impossible to conduct the tests over a sufficiently long period of time to determine the life and dust-holding capacity at these low temperatures. The tests on *Filter A* with oil No. 2 and on *Filter B* indicate that filtering efficiency remains constant over a comparatively wide range of temperatures if there is no overloading caused either by an accelerated rate of feed or by a filter oil too viscous to absorb the dust readily.

As a result of these filter tests conducted at various air temperatures, it may be concluded that viscous filters with comparatively light oils will show no appreciable change in efficiency with temperature when operating between

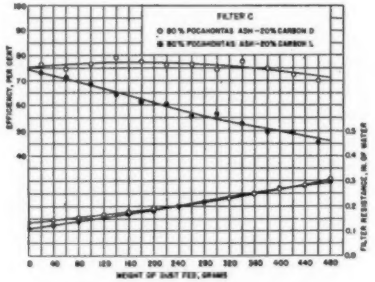

FIG. 8. TYPICAL PERFORMANCE CURVES FOR *Filter C* ON DUST MIXTURES CONTAINING DIFFERENT CARBONS. DUST FEED, 40 GRAMS PER HOUR

0 F and 130 F. From the limited tests conducted it would appear that with all viscous filters there is a tendency toward greater life and dust-holding capacity when operating at high temperatures. Those filter oils showing a marked decrease in viscosity at low temperatures will result in a certain degree of overloading at low temperatures even with test dusts which give no indication of overloading at normal temperatures. Thus it would appear highly probable that any viscous filter subjected to an accelerated test with any dust mixture would show overloading if the temperature were lowered sufficiently.

INVESTIGATIONS OF CARBONS

From the previous investigations it was apparent that one of the principal components of the test dusts causing filter overloading during the accelerated tests was carbon. For this reason it was decided that more extensive investigations should be carried out for the purposes of determining those physical properties resulting in overloading and of finding a standard product which would neither vary in its physical properties from lot to lot nor result in

filter overloading during test. Letters were first written to 16 different manufacturers of carbon blacks and lampblacks, explaining the problem to each of them and asking them to submit samples which, in their opinion, would be the most likely to satisfy the requirements of a carbon to be used in a test dust for air filter ratings. As a result of this correspondence, 14 different carbons were submitted for test by eight different manufacturers. Each of these carbons was given a code letter, as recorded in Table 5 along with the manufacturer's name and designation.

In order to investigate the desirable physical properties of a carbon satisfactory for air filter test purposes, two series of tests were made: one to

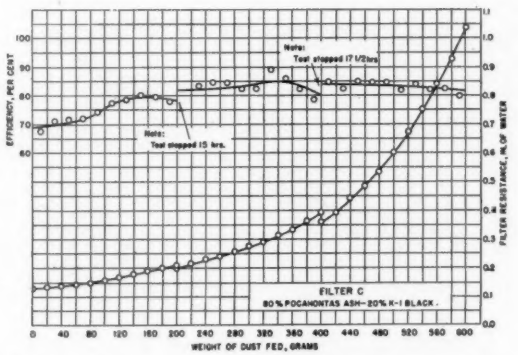

FIG. 9. PERFORMANCE CURVES FOR *Filter C* ON 80 PER CENT POCAHONTAS ASH, 20 PER CENT K-1 BLACK. DUST FEED, 20 GRAMS PER HOUR

determine the physical properties of the carbons and the other to determine the degree to which these carbons resulted in overloading when used in test dust mixtures. The physical properties investigated were the density, the rate of oil absorption, and the weight of oil absorbed. The procedures used in determining each of these properties were as follows:

Density: The jolted density in grams per cubic centimeter was determined for each of the dusts by placing it in a cylindrical container and jolting by hand to maximum density. From measurements of the weight and volume, this jolted density was then calculated.

Weight of Oil Absorbed: Several different standard tests were tried in order to determine the weight of oil absorbed in grams of oil per 100 grams of dust. The test which appeared to be the most easily duplicated and which gave the most consistent results was one developed during these investigations. Two grams of the dust to be tested were weighed and placed in a 250 ml beaker, and refiltered linseed oil with an acid value of 2-4 was then slowly added from a burette. During this process the beaker was rotated vigorously, and from time to time the addition of oil was stopped in order to make sure that sufficient time was being allowed for maximum absorption. The approach of the end point was evident by the complete disappearance of all loose dust and the formation of one large lump or ball. The weight of oil absorbed in grams oil per 100 grams of dust was then calculated from the original weight measurements of the dust and readings of the oil burette.

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Rate of Oil Absorption: It was evident that, in addition to density and the weight of oil absorbed by the test dust, the rate at which the dust absorbed the oil was also an important factor in these investigations of the overloading properties of dusts. A test was therefore devised in which 20 ml of the oil to be used was placed in a 1 in. by 8 in. test tube and ½ gram of the dust under investigation added in a layer above the top surface of the oil. The tube was then placed in a rack and allowed to stand undisturbed until the end point, taken as the complete wetting of the dust by the oil, was reached. Although this test was obviously somewhat affected by the density of the dust and, therefore, the height of the column of dust above the oil, this factor is also encountered during actual filter tests. In order to simulate actual conditions as closely as possible, this rate of absorption test was conducted with the adhesive used on *Filter C* and the No. 2 oil used on *Filter A* (see Table 2).

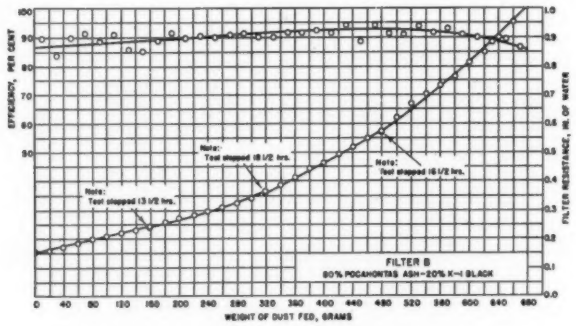

FIG. 10. PERFORMANCE CURVES FOR *Filter B* ON 80 PER CENT POCAHONTAS ASH, 20 PER CENT K-1 BLACK. DUST FEED, 20 GRAMS PER HOUR

To determine the degree to which these carbons were satisfactory for actual test purposes, a series of investigations was made with dust mixtures consisting of 80 per cent by weight of Pocahontas ash screened through a 200 mesh sieve and 20 per cent by weight of the particular carbon under investigation screened through a 100 mesh sieve. *Filter C* was chosen for these tests, as the previous investigations had shown it to be one of the most susceptible to overloading. Filter tests were made at a face air velocity of 300 fpm and at a 40 grams per hour rate of dust feed with each of these dust mixtures. Each test was continued until 480 grams of the dust had been fed to the filter, and from the data collected, the average efficiency, the maximum efficiency, the resistance rise, and the dust-holding capacity were all computed. In addition the drop from maximum to minimum efficiency was recorded and used as a criterion of the degree of overloading.

The results of the tests to determine the physical properties of the different carbons, all screened through 100 mesh, are presented in Table 6. The dust designation is shown in the first column, and the jolted density in grams per cubic centimeter and weight of oil absorbed in grams oil per 100 grams of dust are recorded in the second and third columns. The rates of oil absorption, using the oils from *Filters A* and *C*, are shown in the last two columns.

The jolted densities varied widely, ranging from 0.07 to 0.48 grams per cubic centimeter. The weights of oil absorbed per 100 grams of dust varied from 63 to 796 grams. The rates of oil absorption also varied widely between the different dusts, and, in addition, varied considerably between the two oils, with the adhesive from *Filter C* requiring an average of approximately 180 times as long to absorb the dust as the adhesive from *Filter A*. This slow rate of absorption for the *Filter C* adhesive probably explains the marked tendency for this particular filter to overload under accelerated test.

In addition to the determinations made of the density and oil absorption properties of the different carbons, similar tests were made on Pocahontas ash, Illinois fly ash and Fuller's Earth, all screened through a 200 mesh sieve.

TABLE 5—CARBON BLACKS AND LAMPBLACKS SUBMITTED FOR TEST

CARBON	MANUFACTURER	MANUFACTURER'S DESIGNATION
A	Binney and Smith Co.	Raven No. 15 Carbon Black
B	Binney and Smith Co.	Molacco Carbon Black
C	Binney and Smith Co.	Double Bolted Carbon Dust
D	L. Martin Co.	K–1 Black
E	Samuel Cabot, Inc.	Processed No. 8 Lampblack
F	G. L. Cabot, Inc.	Carbolac 2 Carbon Black
G	G. L. Cabot, Inc.	Elf 30 Carbon Black
H	Monsanto Chemical Co.	No. 10 Lampblack
I	General Atlas Carbon Co.	No. 9010 Black
J	General Atlas Carbon Co.	No. 9011 Black
K	General Atlas Carbon Co.	No. 12509 Black
L	General Atlas Carbon Co.	No. 12511 Black
M	Bihn and Wolff Co.	No. 1012 Lampblack
N	Imperial Oil and Gas Products Co.	Excello Carbon Black

These dusts, in comparison with the carbons, all showed high density and rate of absorption and a low weight of oil absorbed. As none of these dusts had ever shown any tendency toward causing filter overloading, it was considered probable that such properties were necessary for a non-overloading dust.

The results of the tests conducted on *Filter C* with the 80-20 mixtures of Pocahontas ash and the different carbons are shown in Table 7. Typical performance curves for two of these dust mixtures are presented in Fig. 8. These tests show a wide variation in the efficiency drop during the tests for the various dusts, the maximum being 28 per cent and the minimum seven per cent. If it had not been for the varying degrees of overloading experienced with the different test dusts, it appears likely that the average efficiencies for all tests would have been nearly the same, as the maximum efficiencies were all between 70 and 77 per cent with only two exceptions. As would be expected with varying degrees of overloading for the different tests, there were wide variations in the values for dust-holding capacity and for resistance rise.

In order to determine those physical properties of the carbons which had the greatest influence upon filter overloading, rank difference correlation coeffi-

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cients were calculated for each pair of the following factors: rate of absorption (filter oil No. 4), rate of absorption (filter oil No. 2), weight of oil absorbed, jolted density, and efficiency drop. These correlation coefficients are presented in Table 8 and were based upon giving the best rankings to the highest rate of absorption, the lowest weignt of absorption, the lowest efficiency drop, and the highest density. The highest or lowest values were ranked best for each factor, dependent upon which appeared to be preferable as far as preventing filter overloading. In interpreting these results, it would appear that all of the coefficients were of sufficient magnitude to indicate statistically significant results. The correlations between any two of the

TABLE 6—DENSITY AND OIL ABSORPTION TESTS OF DIFFERENT DUSTS

DUST	JOLTED DENSITY, GRAMS PER CU CM	OIL ABSORPTION, GRAMS OIL PER 100 GRAMS DUST	RATE OF OIL ABSORPTION (OIL NO. 4) GRAMS PER HOUR × 10^3	RATE OF OIL ABSORPTION (OIL NO. 2) GRAMS PER HOUR
Carbon *A*	0.22	171	10.6	1.2
Carbon *B*	0.39	72	19.4	6.3
Carbon *C*	0.16	217	3.6	0.6
Carbon *D*	0.48	63	24.6	7.1
Carbon *E*	0.23	140	7.3	2.0
Carbon *F*	0.12	424	1.5	0.5
Carbon *G*	0.17	183	6.7	0.8
Carbon *H*	0.25	116	9.4	1.4
Carbon *I*	0.28	102	10.6	1.6
Carbon *J*	0.21	150	7.3	1.0
Carbon *K*	0.17	203	5.8	0.9
Carbon *L*	0.08	796	0.5	0.9
Carbon *M*	0.07	289	1.2	0.2
Carbon *N*	0.24	164	7.3	1.4
Pocahontas Ash	0.77	66	62.1	20.0
Illinois Fly Ash	0.74	64	85.6	25.8
Fuller's Earth	0.60	80	73.0	12.9

factors, rate of absorption (oil No. 2), rate of absorption (oil No. 4), jolted density, and weight of oil absorbed, were all very high. This indicated that it made little difference whether an oil of low or high viscosity was used in determining the relative rankings of the dust as to absorption, and that a carbon dust which would absorb comparatively small weights of oil would also absorb it rapidly. A dust which had a high density had a high rate of oil absorption, but absorbed only a comparatively small amount of oil. Although the correlations between efficiency drop and rate of absorption, weight of oil absorbed, and density were not as marked as in the other cases, they were, however, of sufficient magnitude to be significant. Furthermore, nine out of the fourteen filter tests using the 80-20 dust mixtures with different carbons resulted in efficiency drops between 19 and 28 percentage units. Because of this clustering, it appears quite probable that if the tests had been conducted over longer periods of time, they would have resulted in more marked differences in efficiency drops between the various dusts and therefore very possibly in higher correlations. Nevertheless, this analysis indicates that

those carbons having the highest rates of oil absorption and the lowest weight of oil absorbed will result in the least amount of overloading. Although these correlations were made only with different blacks, it is interesting to note that Pocahontas ash, Illinois fly ash, and Fuller's Earth have all been used at various times either as components of test dusts or as test dusts themselves without resulting in any filter overloading, and that, as shown in Table 6, these dusts also are all of comparatively high density, high rate of oil absorption, and low weight of oil absorbed. The two carbons, *B* and *D*, which, when mixed with Pocahontas ash resulted in the lowest efficiency drops during the filter tests, both had the highest densities, the lowest weights of oil

TABLE 7—COMPARISON OF PERFORMANCE CHARACTERISTICS ON *Filter C* WITH 80–20 DUST MIXTURES CONTAINING DIFFERENT CARBONS

(*Face Air Velocity* 300 *fpm. Dust rate of feed* 40 *grams per hour. Weight Dust Fed* 480 *grams*)

DUST	AVERAGE EFFICIENCY, PER CENT	RESISTANCE RISE, INCHES OF WATER	DUST COLLECTED IN FILTER, GRAMS	MAXIMUM EFFICIENCY, PER CENT	EFFICIENCY DROP, PERCENTAGE UNITS
A	61.2[a]	0.19[a]	294[a]	70	20[a]
B	72.9	0.14	350	77	7
C	54.1	0.16	260	73	20
D	75.5	0.18	363	77	7
E	68.0	0.25	326	71	10
F	53.4	0.12[a]	256[a]	65	22[a]
G	63.3	0.18	304	70	20
H	70.6	0.23	339	76	11
I	63.1	0.20	303	72	20
J	61.1	0.18	293	73	28
K	59.3	0.18	284	68	24
L	59.0	0.19	283	73	28
M	64.5	0.33	309	74	19
N	60.2	0.15	289	72	27

[a] Approximately 400 grams dust fed and results extrapolated.

absorbed, and the highest rates of oil absorption of any of the blacks tested. These two dusts were respectively Molacco Black and K-1 Black. (See Table 5.)

NON-OVERLOADING DUST MIXTURES

The results of the filter tests of different carbons and of the 80 per cent Pocahontas and 20 per cent carbon mixtures indicated the physical properties of dusts possessing satisfactory non-overloading characteristics for accelerated filter tests. However, in addition to these characteristics, any such dusts satisfactory for a standard mixture must not vary appreciably in physical properties from batch to batch during manufacture. There are probably many different dusts which would meet most of the requirements of a satisfactory mixture; there are also probably many different ratios of these dust

components which would be satisfactory. The overload tests already discussed indicated that one such mixture was that composed of 80 per cent Pocahontas ash and 20 per cent K-1 Black. However, in order to establish fully that this was the case, two additional tests were run with this mixture on *Filters C* and *B* at a rate of dust feed of 20 grams per hour and to a final limiting resistance of 1 in. of water. These tests were run intermittently with time lapses allowed between each eight hours of actual test. In addition eight different four-hour tests were run on *Filter B* using seven different samples of K-1 Black obtained from different production batches at different dates. The rate of dust feed was 20 grams per hour and no other components were added to the test dust.

Table 9 presents the results of the four-hour tests made on *Filter B* with K-1 Black and records the sample designation, the jolted density, the average

TABLE 8—RANK DIFFERENCE CORRELATION COEFFICIENTS BETWEEN EFFICIENCY DROP, WEIGHT OF OIL ABSORBED, RATE OF OIL ABSORPTION AND DUST DENSITY

	WEIGHT OF OIL ABSORBED	JOLTED DENSITY	EFFICIENCY DROP	RATE OF ABSORPTION (OIL NO. 2)
Rate of Absorption (Oil No. 4)	0.93	0.93	0.53	0.84
Rate of Absorption (Oil No. 2)	0.91	0.95	0.50	
Efficiency Drop	0.60	0.52		
Jolted Density	0.96			

efficiency, and the resistance rise. The densities for the different samples varied from 0.428 to 0.464 grams per cubic centimeter and the average efficiencies for the four-hour tests ranged from 75.0 to 88.5 per cent. Much of the variation in efficiency may be attributed to differences in the filters even though of the same brand and manufacture. For example, dust sample *G* was tested on two supposedly identical filters, and yet the average efficiencies for these tests were 75.0 and 81.0 per cent. In view of the results obtained on other carbons with similar tests at a previous date,[6] these results may be considered very satisfactory from the standpoint of duplication.

Figs. 9 and 10 are the performance curves for the tests conducted on *Filters C* and *B* with the 80 per cent Pocahontas, 20 per cent K-1 Black mixture to a limiting resistance of 1 inch of water. Each of these tests was run in 8-hour increments with time lapses as noted on the graphs. The test on *Filter B* showed practically no indication of overloading with only slight changes in resistance and no noticeable changes in efficiency between test periods. The test on *Filter C* indicated a slight overloading as evidenced by changes in resistance and efficiency during the intervening time periods. A

[6] Loc. Cit., Note 1.

similar test run on *Filter C* with this dust mixture at 20 grams per hour but with continuous instead of intermittent testing showed no indication of overloading, although of course slight overloading was probably taking place. In the light of previous research on dust mixtures it appears that regardless of the mixture, a filter utilizing an adhesive as viscous as that on *Filter C* will evidence a slight overloading during accelerated tests. However, a slight overloading apparently has little, if any, effect upon the efficiency and only a small effect upon the life and dust-holding capacity. These tests therefore indicate that the dust mixture consisting of 80 per cent Pocahontas ash,

TABLE 9—COMPARISONS OF BLACK D[a] ON *Filter B*
(*Dust Feed 20 grams per hour. Face Air Velocity 300 fpm*)

TEST NO.	SAMPLE	JOLTED DENSITY, GRAMS PER CU CM	AVERAGE[b] EFFICIENCY, PER CENT	RESISTANCE[b] RISE IN. OF H_2O
1	(a)	0.437	82.7	0.056
2	(b)	0.434	80.9	0.066
3	(c)	0.441	81.0	0.043
4	(d)	0.408	75.5	0.055
5	(e)	0.443	88.5	0.047
6	(*f*)	0.464	81.4	0.044
7	(g)	0.428	81.0	0.052
8	(g)	0.428	75.0	0.064

[a] All same brand and same manufacture but different production batches.
[b] Based on 4-hour average.

screened through 200 mesh, and 20 per cent K-1 Black, screened through 100 mesh, is one mixture which satisfies the requirements as to reproduction of components and as to non-overloading characteristics required of a standard dust mixture.

SUMMARY

In this paper it has been shown that there is apparently no danger of filter overloading during accelerated tests of dry filters, but that with oiled filters overloading may occur if the rate of dust feed is more rapid than the rate with which the adhesive can absorb the dust. If such is the case, overloading may result in fictitious performance characteristics usually showing a decrease in average efficiency and an increase in life and dust-holding capacity over normal performance. Such a condition can be brought about if the filter adhesive is extremely viscous and incapable of absorbing dust rapidly, if the final limiting resistance is arbitrarily placed much beyond that to which the filter may normally be subjected under good operating conditions, or if low density test dusts having a low rate of oil absorption and a high weight of oil absorption are used in the test mixture. Overloading can be detected during accelerated tests either by a drop in efficiency as the test progresses, a sudden change in efficiency and filter resistance after an interruption of

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appreciable duration during a test, or by a change in the performance characteristics of like filters when tested with the same dust mixture but at different rates of feed.

Tests conducted on viscous filters with comparatively light adhesives showed no appreciable change in efficiency with temperature when operating between 0 F and 130 F. However, those filters with oils having a marked decrease in viscosity showed a certain degree of overloading at low temperatures even with a test dust which gave no indication of overloading at normal temperatures. There appeared to be definite tendency toward greater life and dust-holding capacity at high temperature for all viscous filters tested.

As a result of tests conducted on various carbon blacks and lampblacks submitted by different manufacturers, it was found that a dust, to possess satisfactory non-overloading characteristics when used as a component of a filter test dust, should exhibit a high rate of oil absorption, but a low weight of oil absorption. In addition, any dusts satisfactory for use in a standard test mixture must be capable of duplication. One dust mixture tested which satisfied these requirements as to reproduction of components and non-overloading characteristics consisted of 80 per cent Pocahontas ash screened through 200 mesh and 20 per cent K-1 Black screened through 100 mesh.

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No. 1219

AIR CONDITIONING OF BLACKOUT PLANTS

By WALTER A. GRANT,* PHILADELPHIA, PA.

OUTSTANDING in the vitally important prosecution of our war effort is the gigantic expansion in industrial building which has occurred in the last two years. Despite the speed of construction, suitability to the task to be performed has been the guiding consideration in the design of these structures. Most stirring to the public imagination perhaps has been the construction of the enormous factories for the production of aircraft and aircraft engines.

Some confusion exists as to what constitutes a blackout plant. Broadly speaking, every manufacturing plant whether built with windows (Fig. 1) or windowless can be blacked out when the necessary precautions have been incorporated either before or after the building has been completed. Supplementing the term *blackout* therefore, it is desirable to draw the distinction between factories which have windows and those which do not.

The processes for which these many plants have been built are almost infinite in number, but it is a fact that, in general, the technique of designing the heating or the air conditioning is relatively independent of them. The process may, and usually does, determine whether simple heating, or partial or complete air conditioning is to be used. Beyond that, the treatment is largely a function of the building—its shape, size, interior divisions, exterior exposures, structural influences and interferences, population distribution, and other job conditions.

None of these considerations are new or unique, and the technique of heating or air conditioning a factory building is probably the simplest and least troublesome application of the art that the industry has had occasion to face, and it has faced it for a good many years. The really only new element that has entered the picture is the element of *size,* and this will be recognized as the same element which has captivated popular interest and wonder in the aircraft engine and assembly plants. These large special-purpose factories have been built for the mass production of one particular instrument of warfare, and in more ways than one typify the resourcefulness and resolution of the technical brains behind our war effort. The comparison between aircraft assembly and aircraft engine plants is useful and enlightening, since their

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requirements are quite different. Furthermore, as was previously implied, in general the solution of these requirements and the conclusions therefrom are applicable to almost all modern industrial buildings.

HEATING, VENTILATING AND AIR CONDITIONING

A comprehensive task confronts the heating and ventilating engineer. The complete treatment of one of these plants involves most and frequently all of the following functions:

1. Adequate treatment of the factory working spaces is perhaps the largest single item. Whether heating with natural or forced ventilation, or evaporative cooling, or all of the complete air conditioning functions—this part of the project usually involves treatment for the largest number of employees, requires the greatest expenditure of

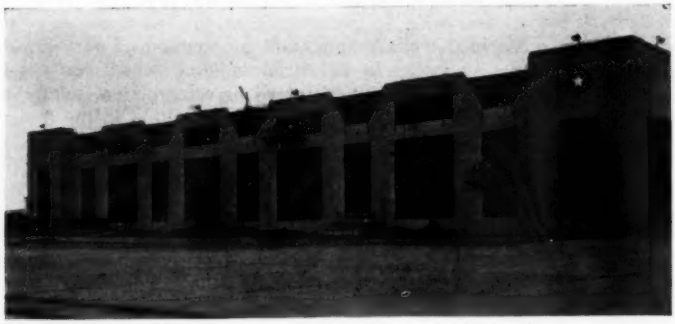

FIG 1. BLACKOUT CURTAINS BEING APPLIED TO PLANT WINDOWS

money in equipment, and uses the most fuel and power to operate. Atmospheric conditions require control in some cases for the benefit of the product, in others for the promotion of human efficiency.

2. Production of comfortable conditions for the office employees is of course necessary. Frequently either a separate office building is part of the project, or a portion of the building is segregated for the management, technical and clerical staffs.

3. Most plants have special-purpose rooms, devoted to testing, precision operations, or storage requiring special conditions of indoor weather.

4. Other miscellaneous spaces such as dead storage for materials, receiving platforms, etc. require simple heating only.

5. Where thousands of people are to be fed, there is always the application of refrigeration and air conditioning for the preservation of food.

6. Certain localized areas such as heat treating rooms, or manufacturing spaces where fuel is burned in large quantities require special precautions to maintain at least tolerable working conditions.

7. There is always a fairly extensive ventilation and exhaust requirement for kitchens, toilets and locker rooms.

8. Finally, special exhaust systems are frequently required for the collection of fumes in pickling, electro-plating, painting, wet cleaning, anodizing operations, and the removal of dust in sandblasting and grinding.

Most of these problems are solved in the conventional manner which has developed and progressed with the art during the last few years. The aspect which has received the focus of most attention is the treatment of the main

manufacturing and assembly areas, and it is toward this aspect that this discussion is devoted.

MAIN FACTORY WORK SPACES

In the modern aircraft assembly and engine plant, these spaces have certain distinguishing characteristics, some of which are relatively new. In the first place the dimensions in all directions exceed anything which has been contemplated in the past. Some of these plants are as long as 4,000 ft in a single building with widths of 500 ft. The engine plants normally have clear ceiling heights under the trusses of from 14 ft to 18 ft, which will be recognized as conventional factory practise. Some of the assembly factories on the other hand run all the way to 45 ft clear height. The span between columns may be very great, and some plants have been built with spans of 300 ft. Partitions are usually few, and when used are generally for the segregation of heavy heat loads from furnaces, paint spraying operations, and the like.

WINDOWLESS BUILDING

There is no hard and fast answer to the question: when should a windowless building be used?—but a general trend is apparent. While it is not necessary to air condition a windowless building, a minimum prerequisite is an adequate system of mechanical ventilation to provide the equivalent of movable sash in a building with sawtooth or monitors. But, from the opposite point of view, where air conditioning is considered *necessary*, the entire operation is *simplified* by making the building windowless. The load is drastically reduced, window condensation is avoided, and the problem of good air distribution, particularly with regard to heating, is eased considerably.

The windowless building is more costly to build. The cost does not occur in the structure since the cost of building walls is approximately that of the windows which are replaced. The additional cost includes air conditioning and a number of other items which good practise usually indicates in a windowless building, such as sound insulation on ceiling and walls to reduce internal noise. Against the increase in investment is balanced a cut in the heating cost alone to about one-third of that for a building with windows, plus an improvement in health and efficiency which, while difficult to evaluate, has been proved to be real and important. The opportunity to save fuel in winter, in view of shortages, should not be overlooked.

WHEN SHOULD A PLANT BE COMPLETELY AIR CONDITIONED?

There are only two major factors to be considered in answering that question. The first is related to improvement in the product, the second involves the effect upon the human being. Where high precision work is being performed, there is almost invariably a rigid requirement for the maintenance of tolerances and this implies a uniform temperature year in and year out. Coupled with this requirement is the paramount importance of excluding dirt, which is almost impossible in a building with windows and without air conditioning. Frequently also perspiration must be avoided in the handling of the precision parts. Where precision manufacturing is required, therefore,

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air conditioning is practically essential, and can definitely be evaluated in lower scrap losses and higher production.

The improvement in human efficiency is a real factor but somewhat intangible because of the difficulty in measuring it. The principal result is the lessening of fatigue, with the accompanying benefits, the reduction of spoilage and accidents. Of real importance also is the reduction in absences from sickness, particularly due to respiratory infections.

Some plant processes generate quantities of heat which produce working conditions that are practically intolerable. While data are not as yet sufficient to prove that such conditions are actually a health hazard, the reduction in the output and productivity of labor is beyond question.

In comparing the two types of aircraft projects, assembly plants and engine plants, it can be concluded that engine plants (Fig. 2) usually require complete air conditioning because of the product. It is a fact that the major engine plants are completely air conditioned throughout—Ford, Packard Wright, Allison and Jacobs. The precision requirement, except in special departments, is not generally present in assembly plants, and practise is about evenly divided as to the importance of providing complete air conditioning for the promotion of human efficiency. Representative of plants which use simple heating and open the windows in summer are Bell, Ford Bomber, Martin, Boeing and the first Consolidated plant. Aircraft assembly plants which have been completely air conditioned include Douglas, Grumman, North American and the two latest Consolidated factories.

METHODS OF HEATING, VENTILATING AND AIR CONDITIONING

Basically there are five possible methods which can be effectively used for the treatment of these plants depending on whether they are to be simply heated or fully conditioned:

1. Heating with radiators.
2. Heating with unit heaters.
3. Central groups of heating and ventilating systems.
4. Central groups of evaporative cooling systems.
5. Central groups of complete air conditioning systems using mechanical refrigeration or cold well water or a combination thereof.

The major trend of practise is definitely to use either heating with unit heaters, opening the windows in summer, or to go to complete year 'round air conditioning. Heating with radiators has been considered impractical for these large plants because of the steam distribution and control problem, and its use is therefore confined to the relatively small plant.

Numerically, plants providing simple heating are employing unit heaters almost exclusively, since this method eliminates all ductwork other than air intakes, and usually results in lowest first cost consistent with excellent results. The projection type using propeller fans appears to be favored for buildings of low and moderate ceiling height, whereas with high ceilings or for applications requiring long blows, the centrifugal fan heat diffuser is favored as would be expected. Direct gas fired heaters have been used in several instances.

There are a few notable exceptions to the use of unit heaters, and these have been in assembly factories where process steam has not been required.

Under such circumstances, direct-firing of oil in central air heating equipment has been of advantage. This method implies one or more centralized duct systems, and a preferred arrangement has utilized under-floor ducts, either metal or concrete, distributing air by risers at columns, or through grilles located in the floor. Central ventilation is also used to take care of excessive concentrated heat loads which are commonly partitioned off from the main space. One practise in this respect is use of the displacement method introducing air near the floor and exhaust at the ceiling with a relatively high air change and with provision for spot cooling of the employees.

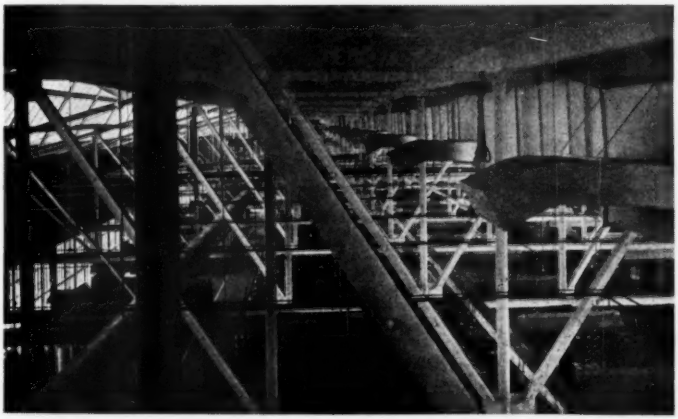

FIG. 2. A VIEW OF PART OF THE DUCT WORK USED FOR DISTRIBUTING CONDITIONED AIR THROUGH A FACTORY

Such design to be successful must be quite flexible, and should take into account the type of occupancy.

An important heating requirement relates to the large doors in assembly plants which must be opened wide in midwinter. Heat diffusers may be used to blow a continuous curtain across from the sides and downward from overhead locations. The central underground duct permits a continuous slot in the floor the length of the door opening. These methods are all satisfactory on calm days, but can hardly be expected to function adequately with high wind.

A modification in recent heating technique is the use of the blackout ventilator and blackout heater, both offsprings of the war. These devices are equally applicable to new and existing buildings, and recognize the obvious fact that mechanical ventilation must be provided when a building with movable sash is blacked out. The ventilators are simply supply or exhaust fans, incorporating a seal or baffle to prevent passage of light, an intake or hood to prevent entrance of rain, means of attachment to the roof or wall, and adequate facility for picking up or distributing the air. The blackout heater is a supply ventilator with heating coil. If factory operation under

blacked-out conditions becomes necessary, such equipment will achieve considerable importance.

Evaporative cooling has not been used to any large extent, except in a few plants for the cooling of spaces subjected to excessive internal heat loads. Apparently the potentialities of this method for producing improved comfort

TABLE 1—COMPARATIVE COMFORT PRODUCED BY VENTILATION, EVAPORATIVE COOLING AND COMPLETE SUMMER AIR CONDITIONING

DIFFERENCE BETWEEN ROOM AND SUPPLY AIR TEMPERATURE (Δt)	CONDITION[a]	OUTSIDE CONDITIONS (F)				INSIDE CONDITIONS (F)			
		DB	WB	Per Cent RH	ET	DB	WB	Per Cent RH	ET
		SIMPLE VENTILATION							
5 F	*A*	95	78	48	84.5	100	79.3	41	86.5
	B	90	70	37	79.4	95	71.5	32	81.5
	C	82	78	84	79.6	87	79.3	72	82.0
10 F	*A*	95	78	48	84.5	105	80.4	36	88.5
	B	90	70	37	79.4	100	72.9	28	83.5
	C	82	78	84	79.6	92	80.4	62	84.5
		EVAPORATIVE COOLING (*with 95 Per Cent Saturation efficiency*)							
10 F	*A*	95	78	48	84.5	89	80.4	70	83.5
	B	90	70	37	79.4	81	72.9	68.5	76.5
	C	82	78	84	79.6	88.2	80.4	72	83.5
15 F	*A*	95	78	48	84.5	94	81.7	59.5	85.5
	B	90	70	37	79.4	86	74.2	58	79.5
	C	82	78	84	79.6	93.2	81.7	61.5	85.5
		AIR CONDITIONING WITH DEHUMIDIFICATION							
Any	*A*	95	78	48	84.5	80	66.7	50	74
	B	90	70	37	79.4	80	66.7	50	74
	C	82	78	84	79.6	80	66.7	50	74

[a] Condition *A* is a design maximum; *B* is a typical sunny dry day; *C* a typical cloudy muggy day.

have been overlooked for many plants, particularly those with heavy heat concentrations. Evaporative cooling can by no means be construed as giving comfortable conditions all summer long. But it is capable of giving results equal to complete air conditioning during the fall, winter and spring seasons, and results definitely superior to straight ventilation, with lower air changes, in the summer time (Table 1). The chief reason for considering evaporative cooling now lies in the non-use of many of the critical materials required for complete summer air conditioning.

AIR CONDITIONING TECHNIQUE

Of considerable interest to the engineer is the question of design heating and cooling loads for these large plants. Heating loads are comparatively small in a windowless factory, since roof and usually walls are insulated, and heating is largely a question of tempering ventilation air in the coldest weather. Some of these plants have an internal load requiring cooling (by outside air) down to external temperatures of 10 or 20 F. This is not at all true of factories with sash in sawtooth or monitor, where the heating load becomes a major consideration, and is largely a function of the amount of glass used.

With respect to summer cooling, the external load on the windowless plant is comparatively small, due again to roof insulation. The internal load is highly variable, but is mainly a function of the horsepower of machines in operation. Electric lights and people are additional sources of room load. Table 2 presents tabulated values of load which may be considered to cover the average range of windowless building application. *Section A* shows a moderately low concentration of people and electric lighting; in *Section B* these items are moderately heavy.

No truly centralized system of ventilation or air conditioning is feasible because of the vast dimensions of the spaces to be conditioned. A single central system would involve ducts of impractical length and section dimensions, and apparatus of excessive size. At the present time, 400 ft has been considered about the maximum for the length of an air duct, and even here uniform distribution is somewhat of a problem. The practical considerations limiting duct size are the problem of running the duct between steel members of trusses; the progressively greater difficulty in stiffening and bracing ducts as the size increases; the greater load imposed on trusses and building columns by trunk ducts; the obstruction to lighting; and the greater difficulty of balancing air distribution as size increases. The limits on size of fans, coils and washers are those imposed by available headroom, load concentration and the departure from standard equipment available on the market.

All of these limitations are economic rather than technical, and while systems of almost any size can be built, the economic maximum seems to be at about 100,000 cfm in one system, with preferred practise falling between 40,000 and 65,000 cfm. This latter range appears to result in the lowest overall cost when related to the building structure. Although small units are feasible, the problem is quite different from unit heaters and heat diffusers where there are no ducts, piping connections are simple, and maintenance is at a minimum. The experience of the industry has pretty well proved, over a good many years, that a multiplicity of small conditioning equipment is economically justified only where space for large apparatus and ducts is at a premium or not available at all, or where a fine degree of zoning is essential. Neither situation is a factor in the blackout plant.

The preferred arrangement of air conditioning equipment is off the floor, with location at appropriate intervals either on a mezzanine extending along one or both of the long sides of the building, or up in the trusses, or on mezzanine platforms distributed at proper locations throughout the work area. While the latter arrangement does cut down somewhat on the size of ducts, it may be an objectionable obstruction to ceiling height in many plants.

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TABLE 2—HEAT LOADS AND AIR QUANTITIES FOR WINDOWLESS BUILDINGS

A—PEOPLE CONCENTRATION: 160 SQ FT PER PERSON. LIGHTS: 1.5 WATTS PER SQ FT

Hp/1000 sq ft.....	0.5	1.0	1.5	2.0	3.0	5.0	7.5	10.0	15.0
Rm. Sens Heat Btu/hr/sq ft....	13.0	14.5	16.0	17.5	20.5	26.5	34.0	41.5	56.5
Grand Total Heat Btu/hr/sq ft....	17.5	19.0	20.5	22.0	25.0	31.0	38.5	46.0	61.0
Δt = TEMP. DIFF. BETWEEN ROOM AND SUPPLY AIR (F)	CFM/1000 SQ FT (UPPER FIGURE), AND AIR CHANGE, MINUTES (LOWER FIGURE).								
5 F	2400 8.3	2700 7.4	3000 6.7	3250 6.2	3800 5.3	4900 4.1	6300 3.2	7700 2.6	10400 1.9
10 F	1200 16.7	1350 14.8	1500 13.3	1620 12.4	1900 10.5	2450 8.2	3150 6.4	3850 5.2	5200 3.8
15 F	800 25	900 22	1000 20	1080 18.5	1270 15.7	1630 12.3	2100 9.5	2600 7.7	3500 5.7
20 F	600 33	680 29	750 27	810 25	950 21	1220 16.5	1570 12.7	1920 10.4	2600 7.7
25 F	480 42	540 37	600 33	650 31	760 26	980 20	1260 16	1540 13	2080 9.7

B—PEOPLE CONCENTRATION: 80 SQ FT PER PERSON. LIGHTS: 3 WATTS PER SQ FT

Hp/1000 sq ft.....	0.5	1.0	1.5	2.0	3.0	5.0	7.5	10.0	15.0
Rm. Sens. Heat Btu/hr/sq ft....	19.5	21.0	22.5	24.0	27.0	33.0	40.5	48.0	63.0
Grand Total Heat Btu/hr/sq ft....	28.5	30.0	31.5	33.0	36.0	42.0	49.5	57.0	72.0
Δt = TEMP. DIFF. BETWEEN ROOM AND SUPPLY AIR (F)	CFM/1000 SQ FT (UPPER FIGURE), AND AIR CHANGE, MINUTES (LOWER FIGURE).								
5 F	3600 5.5	3900 5.1	4200 4.8	4500 4.5	5000 4.0	6100 3.3	7500 2.7	8900 2.3	11600 1.7
10 F	1800 11.1	1950 10.2	2100 9.5	2250 8.9	2500 8.0	3050 6.6	3750 5.3	4450 4.5	5800 3.5
15 F	1200 16.7	1300 15.4	1400 14.3	1500 13.3	1650 12.1	2050 9.8	2500 8.0	2950 6.8	3900 5.1
20 F	900 22.0	970 21.0	1040 19.3	1110 18.0	1250 16.0	1520 13.2	1870 10.7	2220 9.0	2900 6.9
25 F	720 28.0	780 26.0	830 24.0	900 22.0	1000 20.0	1220 16.4	1500 13.3	1780 11.2	2330 8.6

NOTES: (1) Outside conditions: 95 F DB, 78 F WB; inside: 80 F and 50 per cent.
(2) Roof assumed insulated.
(3) One horsepower taken as 3000 Btu per hour, to allow for motor losses dissipated to room.
(4) Air change based on 20 ft ceiling height. For other heights, correct in proportion.
(5) Grand total heat includes ventilation load of 10 cfm per person.

Most of the plants already constructed employ finned cooling coils, and a few of them have used a single coil served by a central water circulating system for both heating and cooling, at considerable economy in the cost of surface. Direct expansion is used in some cases where the condensing equipment is located nearby. With the national effort to conserve copper, it is probable that spray dehumidifiers may be considered in some of the future plants to avoid the use of copper coils.

Ducts are usually straight single runs, with air velocities up to 2,000 fpm or higher. Ceiling type diffuser outlets of large capacity are in general use, but a real problem of air distribution is involved in winter with ceiling heights greater than 30 ft. The conventional diffusion outlet ejects little or no air with the downward velocity necessary to produce adequate mixing during the heating season. Progress is being made by several designers, however, in developing diffusers with manual or automatic direction and velocity control for this specific purpose. This distribution problem is more difficult for the building with windows, where heat losses from glass result in a relatively high temperature differential between supply air and room. With a windowless plant this becomes a problem only on the very coldest days, since the internal load is normally sufficient for most of the heating.

Air cleanliness is of paramount importance to aircraft engine plants and any other factories performing high precision machine work. Electric precipitating equipment is a favored method for this service.

REFRIGERATION EQUIPMENT

These vast projects require huge tonnages of mechanical refrigeration. There has been much discussion in the technical press concerning the wisdom of centralizing this equipment in one location as opposed to decentralizing it in comparatively small units scattered throughout the plant. The major claims by the proponents of decentralized refrigeration are the minimized effects of bombing or sabotage, the saving of floor space, the ability to stock identical spare parts, improved flexibility of zoning and saving in duct work.

Since the air conditioning equipment must be decentralized anyway, as explained previously, the claims for flexibility of zoning and saving in ducts are hardly valid. The stocking of standardized spare parts would hardly be expected to offset the excessive maintenance of a large number of such equipments. The saving of floor space by location up among the trusses or on the roof is probably more than offset by the advantages of having the mechanical refrigerating equipment centralized in one or two spots. However, there remains the foremost and major contention of the proponents of decentralization—that is, avoidance of the effects of bombing and sabotage. This claim requires a careful consideration of all the factors involved.

No mechanical equipment can withstand a direct bomb hit without being extensively damaged. If the refrigeration equipment is distributed among a large number of small systems all over the plant, a bomb hit anywhere is probably going to wreck the unit in the immediate vicinity of the hit. Since these systems are direct expansion, the blast effect of the explosion will unquestionably open up the refrigerant piping in all of the system in the vicinity of the hit, the radius depending upon the size of the bomb. Inasmuch as fire will almost inevitably result, consideration must be given to the effect

upon workmen and fire fighters of the decomposition products of the refrigerant when exposed in considerable quantities to open flames. Outside of this factor of impeding fire fighting and possibly contributing to a panic, it is certainly evident that such a hit will not merely demolish a single unit but in all probability will put a good many more units out of service.

The centralized refrigeration plant is not subject to these objections. It too will be put out of service with a direct hit, but the statistical odds against a direct hit are very large. The refrigeration plant can be put underground and located in its own equipment room, and therefore can be made relatively immune to the blast effect of any nearby explosions. Any decomposition products caused by fire in the refrigeration room will be confined therein and not spread into the working spaces. The use of chilled water instead of direct expanding refrigerant eliminates the hazard in the working spaces, and furthermore the water piping may run largely underground and if hit or damaged can be fairly promptly repaired.

In addition to these important safety features of the central refrigerating plant, it is obvious that from the viewpoint of operation and maintenance the problem is vastly simplified over that required with a multitude of small equipments. With the air handling equipment necessarily decentralized, the flexibility in shifting load from zone to zone as required is even greater than where the refrigerating capacity is a fixed quantity for each zone.

Speaking broadly, decentralization of refrigeration should bear some relation to the decentralization of buildings and of the mechanical equipment therein. This must follow the master plan of the government concerning the construction of plants within the effective bombing range, which is generally considered within 300 miles of the east or west coastlines. Some of the plants mentioned are thus located and here decentralization of plant is being effected by separation of buildings by at least several hundred feet and in some cases by several thousand feet. In plants where the bombing factor is considered one for such vital precautions, a valuable guide for the refrigeration can be obtained from the practise with regard to the boiler plants. Obviously heating is more important than cooling in order to insure continuous production and if it becomes necessary to decentralize boiler equipment it can be logically assumed that equivalent decentralization of the refrigeration is in order.

The outstanding example of decentralization is at Douglas where there are 11 separate buildings of which four are the general manufacturing plants. The total project includes 23 boiler rooms and 68 self-contained direct expansion air conditioning and refrigerating systems averaging 60 tons, for a combined total of 4,026 tons of refrigeration. Such extensive complication has not been found necessary in most other large projects that are fully air conditioned. Packard with 1,000 tons, Allison 2,550 tons, Ford 6,000 tons, Wright 6,000 tons, Consolidated with 7,000 tons, and others have all used centralized refrigeration plants and have discarded as impractical any idea of complete decentralization of boilers and refrigeration.

Probably the soundest answer, if it is not desired to put all eggs in a single basket, is to divide both boiler and refrigeration equipment into two or possibly three plants at opposite ends of the project. At slightly greater expense the steam and chilled water distributing systems can be interconnected to provide at least partial heating or conditioning for the entire factory in case one of the plants is put out of service.

While cooling towers are conventionally used in most of the air conditioned blackout plants not provided with adequate river or well water, it is of interest to note that a few factories have adopted roof spraying or roof ponds for the dual purpose of cooling the condenser water while simultaneously reducing the solar transmission through the roof to a negligible value.

SUMMARY

In conclusion it can be stated that certain trends affecting our industry are evident as a result of the enormous expansion of industrial plants during this war emergency. It is true that there is nothing new or revolutionary in the engineering and design technique employed in heating or air conditioning a blackout plant. Outside of the problems arising out of building size, methods follow the sound practises built up over a good many years. Of considerably greater importance than the application technique is the extent to which air conditioning has been applied.

Not only is air conditioning being used in sizes hitherto unthought of, more significantly, air conditioning is being employed in more different kinds of plants, and for more different processes than ever before. No longer is it confined to industries where production without it is impossible. It is being employed in plants where production without conditioning has been feasible in the past. There is only one reason for its use, but that reason is so basic, so vital and so compelling that all other considerations are thrown completely to the wind. That reason is the paramount necessity for speed, speed and more speed; the importance in reducing rejects and spoilage to an absolute minimum, and the necessity for keeping every man and every machine at peak production efficiency until the threat to our national existence has been overcome.

This trend cannot help exerting a profound effect upon our industry in the years to come. That is important in the long-term view. It is wholly secondary at the moment. For while true it may be that our armament program has advanced air conditioning—the vital factor right now is that air conditioning is recognized as an essential industry advancing our armament program.

DISCUSSION

W. L. FLEISHER, New York, N. Y.: It was pleasant to note that the author referred to my activities in connection with evaporative cooling for industrial plants, but I also want to correct a misunderstanding which has been prevalent for a number of years—that I have been advocating evaporative cooling for maximum comfort in all types of buildings and under all circumstances.

In December 1936, I presented a paper before the *American Society of Refrigerating Engineers*, entitled "Evaporative Cooling for Comfort." This paper having been presented before a refrigerating society, created a great amount of comment and criticism, the assumption being that if I presented such a paper before a refrigerating group, I was advocating the dis-use of refrigeration for comfort in lieu of evaporative cooling. In that paper and in many papers and talks since, I have stated and have shown by figures and by other references that under many conditions of outside wet-bulb, comfort conditions could be obtained by proper evaporative means, inde-

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pendent of refrigeration, but that when the outside wet-bulb reached 70 deg comfort could not very well be obtained. I pointed out that the work that was carried on in the ASHVE Research Laboratory under the Technical Advisory Committee on Air Conditioning in Industry, showed that safety for the worker, from a physiological standpoint, could be maintained with a properly designed evaporative cooling system, whereas with straight ventilating or air change, with outside conditions of 95 deg dry-bulb, a condition would arise inside the building of approximately 105 deg and 88 or 89 deg ET, possibly more, which conditions would definitely be unworkable or unproductive and would result in inefficient production, not justified by the fact that with evaporative cooling no more fundamental or basic material would be required than with straight ventilating and that conditions below 83 deg ET temperature could be maintained. If the cooler air were obtained by evaporation and directed with a certain velocity at the workers, they would have a feeling of comfort which would be definitely beneficial to their production of material.

I have presented these facts to the War Production Board in Washington and asked that they define accurately what they meant by air conditioning; that certainly for blackout buildings in industry, ventilation was required and that evaporative cooling was not air conditioning, but simply a refined method of ventilation, which, by the very results obtained, could not be construed as air conditioning, as covered by the definition of the ASHVE. I have been informed that the War Production Board's Air Conditioning and Refrigeration Division was giving very careful consideration to my request and that the likelihood of having evaporative cooling under these circumstances defined as ventilation and not refrigeration might be expected.

I wish to compliment the author on the very careful analysis he had made and again assure him of my appreciation of his reference to the great amount of work that I had done to make this type of ventilation or conditioning available to the manufacturer and the general public.

No. 1220

HEAT LOSS STUDIES IN FOUR IDENTICAL BUILDINGS TO DETERMINE THE EFFECT OF INSULATION

By D. B. Anderson,* St. Paul, Minn.

Introduction

THE value of insulation in residential construction (its ability to save fuel and increase comfort) is usually predetermined by heat loss calculations, taking into consideration the theoretical efficiency of the construction with and without insulation. Because there are so many avenues by which heat can escape from a dwelling (walls, windows, ceiling, floor, infiltration and complications of these involving sun and wind), all of which may vary with each different type of house, ordinary methods of calculating heat loss are not necessarily exact. Consequently, claims for the effectiveness of insulation, although they may be made in good faith, will vary considerably. To substantiate or correct the present methods of computation, some tests have been made in the field involving one or perhaps two houses. However, much field work is needed to provide information on this subject.

In order to further increase scientific knowledge, this paper presents the results found in four typical dwellings, each identical to the other except as to degree of insulation. These dwellings were exposed during the heating season to a wide range of weather conditions. It is hoped that these tests will be considered a link in the chain of scientific information which is continually being accumulated in an effort to increase the knowledge on this important subject.

Object

The principal objects of this investigation were:

1. To determine the actual savings in fuel which can be effected by installing various thicknesses of insulation in identical dwellings of typical construction exposed to a wide range of weather conditions.
2. To compare the actual savings with calculated savings.
3. To determine the effect which various thicknesses of insulation have on:
   (*a*) wall and ceiling surface temperatures
   (*b*) temperature gradient from floor to ceiling
   (*c*) attic air temperature.
   (*d*) basement temperature.

* Manager, Sales Engineering Dept., Wood Conversion Co. Associate Member of ASHVE.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, St. Paul, Minn., June, 1942.

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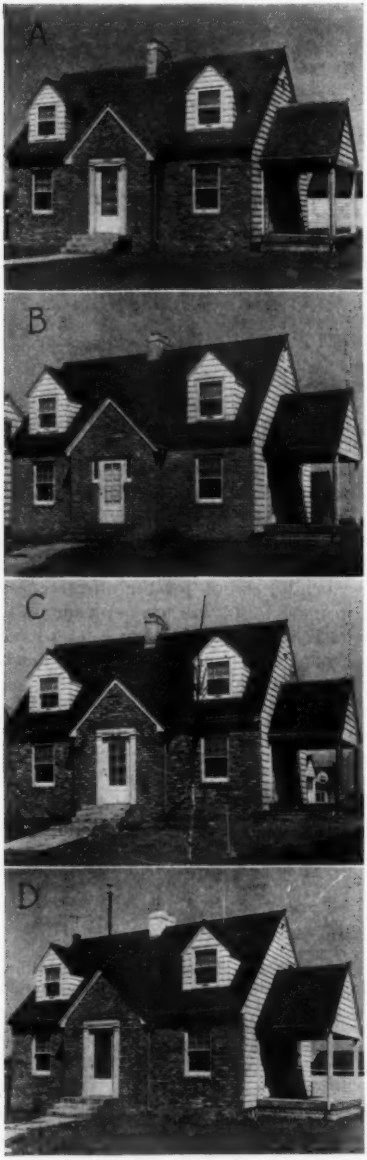

FIG. 1. FOUR TEST HOUSES

DESCRIPTION OF TEST HOUSES

Four houses were involved in this study designated as *A, B, C* and *D,* and shown in Fig. 1. All houses were located within an area of four city blocks, near the north city limits of St. Paul, Minn. In selecting the location for each of the test houses, the relative position of other buildings in the vicinity was considered in order that, as far as practicable, each test house would have equal exposure to sun and wind. The plot on which the houses were constructed was fairly level, and all houses were facing south.

The test houses were 1½-story structures of standard frame construction. With the exception of the degree of insulation, all houses were identical. The plan from which all houses were erected and dimensions of each are shown in Fig. 2. The wall section consisted of rough sand finish plaster on plaster board lath, 2 x 4-in. studs, wood sheathing, building paper, and wood shingles exposed 8 in. and applied in single course. On the first floor front wall only, brick veneer construction was used.

A shed dormer across the entire width of the second floor at the rear, together with the two dormers in front, provided a second floor area nearly as large as the first floor area. The ceiling section consisted of rough sand finish plaster on plaster board lath, applied directly to the underside of joists and rafters. The attic was not floored and a louvered opening in each gable provided means for venting the attic space. The roof section consisted of wood sheathing laid tight, asphalt felt paper with wood shingles exposed 5 in. to the weather. First floor construction consisted of 13/16 in. oak floor, building paper and 1 in. wood subfloor, applied to floor joists. Basement walls were of 12-in. concrete blocks, exposed approximately

12 in. above grade level, and the basement floor consisted of a 4-in. concrete slab.

All windows in the first and second floors were factory weatherstripped and equipped with storm sash and the basement windows were equipped with storm sash but not weatherstripped. Exterior doors were weatherstripped and equipped with storm doors. The finished ceiling heights were as follows: basement 6 ft 11 in. to underside of ceiling joists, first floor 8 ft 2 in., 2nd floor 7 ft 10 in.

Heat was provided by a forced warm-air oil burning furnace having a manufacturer's rated hourly output of 85,000 Btu and the oil burner was of the

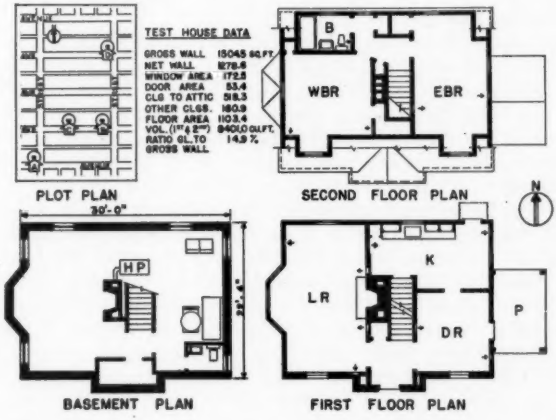

FIG. 2. PLOT AND FLOOR PLAN OF THE FOUR TEST HOUSES

vaporizing type. The primary and secondary heating surfaces of the furnace totaled 42.5 sq ft. The heating plant was equipped with a centrifugal blower, belt driven by a two-speed motor. The furnace casing was provided with an inner liner spaced approximately 1 in. from the exterior casing.

The plant was designed to operate automatically at three stages of fire, as determined by the room thermostat; namely, high, low and pilot. The high and low rates of oil input were determined by the position of two valves, which were actuated by electrically heated bi-metal warp bars. The pilot oil flow was determined by a small metering pump, consisting of a low speed synchronous motor operating a dipper in an oil reservoir. The rate of oil flow at each stage of fire was adjustable. The rate of oil input as maintained in these tests was 0.80 gal per hour at high fire, 0.20 gal per hour at low fire, and 0.04 gal per hour at pilot flame.

On high fire the blower operated at high speed, on low fire and pilot flame the blower operated at low speed. Blower speed was controlled by a thermal relay, actuated by the thermostat circuit so as to switch the blower to full

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speed in proper sequence with high fire operation. In these tests the blower was set to deliver approximately 1000 cfm at high speed with a register velocity of approximately 375 fpm; at low speed the delivery was approximately 500 cfm with a register velocity of 175 fpm. Combustion air to the burner was supplied by a small centrifugal blower attached directly to the shaft of the main blower motor. An adjustable damper was provided in the duct leading from the combustion air blower to the burner which permitted the adjustment of the proper ratio of air to oil input at high fire. With the combustion air blower driven directly by the motor, its delivery was reduced automatically when the main blower operated on low speed. The design was such that an excess amount of air was supplied to the burner under this condition in order to avoid chimney condensation with the low stack temperature encountered during low fire operation. The location of the heating plant is shown in Fig. 2.

A draft regulator of the pressure balancing type was installed in the stack leading from heating plant to chimney. The duct design was identical in all houses and the warm-air supply and return registers were located as shown in Fig. 2. No humidification was provided. All registers were of the baseboard type, with air flow deflected 15 deg downward and the return air was filtered. The room thermostat was located on an inside wall of the dining room at a height of 49 in. above the floor, or one-half level, floor to ceiling.

House A was uninsulated; the three other identical houses were insulated with blanket insulation in exterior walls and second floor ceiling as follows: In *House B* the insulation was 0.9 in. thick, in *House C* the insulation was 1.7 in. thick, and in *House D* the insulation was 2.3 in. thick. The insulating mat consisted of felted wood fibers enclosed between continuous liners of asphalt saturated and coated paper. Insulation in *Houses C* and *D* was divided by intermediate liners, separating the total thickness into two individual layers; thus air circulation within the insulating material itself was reduced to a minimum. The conductivity k factor of the insulation was 0.25 Btu at a mean temperature of 70 F. The edges of each strip of insulation were stapled to the inside face of the framing members by means of a spacing flange, which automatically offset the insulation from the inside face of framing and provided an additional air space. The ends of each strip of insulation were fastened to plates or headers. In insulating the second floor ceiling, continuous strips of insulation were placed from the inside—between the rafters of the sloping ceiling, across the ceiling joists, and down the rafters of the opposite ceiling slope. Insulation was applied in accordance with manufacturer's specifications.

The test houses were constructed during the late summer of 1941; they were completed about October 1. Craftsmen employed were of average skill, and as a result, the workmanship on the completed jobs was representative of that found in modern residential construction.

The houses were unoccupied during the tests, for only in this way could the number of times they were entered be definitely controlled. To further regulate the amount of outside air admitted, a definite schedule of opening and closing windows was established as described later in this report. The houses were unfurnished. All first and second floor windows were equipped with shades, which were kept in half drawn position at all times.

DESCRIPTION OF TESTING EQUIPMENT

Temperatures were measured with copper-constantan thermocouples at 43 locations in each of the houses, as described on the temperature record sheet shown in Fig. 3. Readings were taken with a portable direct reading potentiometer as shown in Fig. 4, having a range of — 40 to 160 F. The thermocouples and instrument were checked against a standard thermometer. Essentially, the temperatures measured were air temperatures inside and outside, and surface temperatures on the inside of the construction. Air temperature in each room was measured by a thermocouple located at the center of the room at the one-half level, from floor to ceiling. In the living room and west bedroom additional thermocouples were provided at distances of 1 in. and 30 in. above the floor and 1 in. below the ceiling. Thermocouples at the 30 in. and one-half level positions were shielded with heavy paper cylinders 15 in. in diameter and 9 in. in height, covered on the outside with reflective foil.

All wall surface temperatures were measured with thermocouples located at the one-half level, from floor to ceiling. Temperatures were measured both midway between as well as directly over the studs, the exact position of all thermocouples with respect to framing members having been determined before interior finish was installed. Temperatures were also measured with thermocouples located as follows: Inside glass surface of the north and south windows of the living room (Fig. 8); surface of north and south ceiling slopes in east bedroom; flat ceiling surface in both bedrooms; floor and ceiling surfaces of the living room at *Houses A* and *D* only; floor surface in basement; floor surface of the west bedroom at *Houses A* and *D* only; air in attic at two positions, at distances of 7 ft from each gable (15 ft apart) and midway between second floor ceiling and roof ridge; inside surface of roof boards in attic space (two positions on each slope) and outside air 18 in. from the north wall and 13 ft above the grade level (shielded thermocouple). All thermocouples on the ceiling surfaces were located midway between the framing members.

Fuel oil consumption was determined by weight as shown in Fig. 5. The weighing tank which was connected to the feed line with a flexible hose was located on a platform scale. When required, oil was admitted to the weighing tank from the main supply tank. As a safety measure, two valves were provided in the supply line. When oil was being added to the weighing tank, the valve in the feed line from the tank to the heating plant was closed and the furnace requirements were taken from a reserve of oil held in the cup of the gage ordinarily used for checking the oil flow.

Self-starting electric clocks were installed to time the periods during which the heating plant operated at the various stages of fire. The current to these clocks was controlled by mercury switches, actuated by the position of the oil supply valves. Meters were installed to measure the electric current consumption of the furnaces.

At *House D* only, the following equipment was installed: A maximum-minimum thermometer near the rear entrance on the north wall. An electric cup anemometer for measuring the wind velocity at a point 6 ft above the roof ridge. A weather vane for determining the wind direction is shown in Fig. 1.

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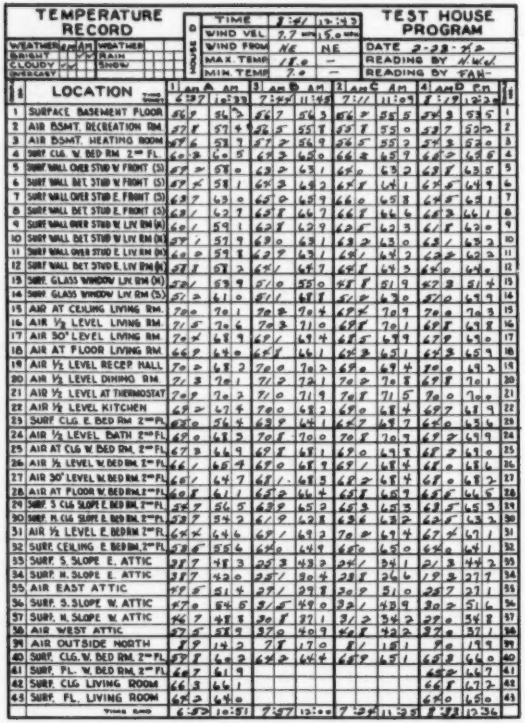

TEMPERATURE RECORD

TEST HOUSE PROGRAM

TIME	8 AM	12 NOON
WIND VEL	2.7 MPH	15.0 MPH
WIND FROM	NE	NE
MAX. TEMP	18.0	—
MIN. TEMP	7.0	—

WEATHER: 8 AM — BRIGHT; CLOUDY; OVERCAST. WEATHER — RAIN; SNOW.

DATE 2-28-42

READING BY H.W.J

READING BY FAN

#	LOCATION	11 AM A	2 PM A	3 AM B	3 PM B	4 AM C	4 PM C	5 AM D	5 PM D	#
	TIME OBS.	6:57	10:33	7:44	11:48	7:11	11:09	8:19	12:23	
1	SURFACE BASEMENT FLOOR	56.9	56.3	56.7	56.3	56.2	56.5	56.3	56.5	1
2	AIR BSMT. RECREATION RM.	57.8	57.4	56.5	55.8	55.8	55.0	55.7	55.2	2
3	AIR BSMT. HEATING ROOM	58.6	58.7	57.2	56.9	56.5	55.2	54.8	55.0	3
4	SURF CLG. W. BED RM. 2ND FL.	60.3	60.5	64.3	64.0	66.2	66.9	66.0	66.6	4
5	SURF WALL OVER STUD W. FRONT (S)	57.2	58.0	63.2	63.1	65.0	63.2	63.8	63.5	5
6	SURF WALL BET. STUD W. FRONT (S)	57.6	58.1	64.3	64.2	65.8	64.1	64.5	64.9	6
7	SURF WALL OVER STUD E. FRONT (S)	58.7	60.0	66.0	66.9	66.0	64.8	64.8	64.1	7
8	SURF WALL BET. STUD E. FRONT (S)	58.1	60.7	66.8	66.7	66.8	66.6	66.3	66.1	8
9	SURF WALL OVER STUD W. LIV RM (W)	60.1	59.1	62.8	62.9	62.5	62.3	61.8	62.0	9
10	SURF WALL BET STUD W. LIV RM (W)	59.1	57.9	63.0	63.1	63.2	63.0	63.1	63.2	10
11	SURF WALL OVER STUD E. LIV RM (W)	60.0	59.8	62.9	63.1	63.1	64.1	63.2	62.2	11
12	SURF WALL BET STUD E. LIV RM (W)	58.8	58.2	64.1	64.7	64.8	64.3	64.0	64.6	12
13	SURF. GLASS WINDOW LIV RM (W)	58.1	58.9	61.0	56.0	58.8	51.9	52.3	51.4	13
14	SURF GLASS WINDOW LIV RM (S)	61.2	61.0	61.1	61.8	61.2	63.0	62.0	63.9	14
15	AIR AT CEILING LIVING RM.	70.0	70.1	70.8	70.4	69.4	70.9	70.0	70.3	15
16	AIR ½ LEVEL LIVING RM.	71.5	70.6	70.3	71.0	68.8	70.1	68.8	69.8	16
17	AIR 30° LEVEL LIVING RM.	70.4	68.9	68.1	69.4	68.5	68.9	67.9	68.0	17
18	AIR AT FLOOR LIVING RM.	66.5	64.0	66.5	66.1	65.3	65.1	65.3	65.9	18
19	AIR ½ LEVEL RECRD HALL	70.0	68.2	70.0	70.2	69.0	69.4	70.0	69.3	19
20	AIR ½ LEVEL DINING RM.	71.3	70.1	71.2	72.1	70.0	70.8	69.8	70.1	20
21	AIR ½ LEVEL AT THERMOSTAT	70.0	70.2	71.0	71.9	70.8	71.5	70.0	70.0	21
22	AIR ½ LEVEL KITCHEN	68.2	67.4	70.0	68.2	69.0	68.4	69.7	68.9	22
23	SURF CLG E. BED RM 2ND FL.	63.0	56.4	63.9	64.1	66.7	64.7	64.0	63.6	23
24	AIR ½ LEVEL BATH 2ND FL.	69.0	68.3	70.5	70.0	70.5	70.3	69.2	69.9	24
25	AIR AT CLG W BED RM. 2ND FL.	67.3	66.9	69.5	68.1	69.0	69.3	68.2	69.0	25
26	AIR ½ LEVEL W. BED RM. 2ND FL.	66.1	66.4	67.0	66.9	69.1	68.4	68.0	68.6	26
27	AIR 30° LEVEL W. BED RM. 2ND FL.	66.1	64.7	66.1	68.3	68.2	67.4	67.0	68.2	27
28	AIR AT FLOOR W. BED RM. 2ND FL.	60.8	61.1	63.2	66.4	66.8	65.7	65.5	66.5	28
29	SURF. S CLG SLOPE E. BED RM. 2ND FL.	58.7	56.5	63.9	65.2	66.3	63.3	63.5	63.3	29
30	SURF. N. CLG SLOPE E. BED RM. 2ND FL.	58.7	54.2	61.9	62.8	63.6	63.2	62.5	63.3	30
31	AIR ½ LEVEL E. BEDRM 2ND FL.	64.4	64.6	69.1	69.3	70.0	69.4	67.4	67.1	31
32	SURF. CEILING E. BED RM. 2ND FL.	58.6	56.6	64.0	64.9	66.0	65.0	64.0	64.1	32
33	SURF. S. SLOPE E. ATTIC	38.7	48.3	50.3	48.3	50.1	54.1	51.3	44.2	33
34	SURF. N. SLOPE E. ATTIC	38.7	42.0	38.1	38.4	38.8	36.6	39.3	37.7	34
35	AIR EAST ATTIC	49.6	51.4	47.1	49.8	50.0	50.1	48.7	47.1	35
36	SURF. S. SLOPE W. ATTIC	47.0	54.6	51.5	49.0	48.1	48.9	50.2	51.6	36
37	SURF. N. SLOPE W. ATTIC	46.7	48.8	50.8	47.1	48.3	48.2	49.0	49.8	37
38	AIR WEST ATTIC	47.5	48.9	47.0	46.9	46.8	46.2	47.0	47.1	38
39	AIR OUTSIDE NORTH	8.9	14.2	7.8	17.0	8.1	18.1	9.0	15.9	39
40	SURF. CLG. W. BED RM. 2ND FL.	58.8	60.2	64.2	66.4	66.8	65.1	66.8	66.0	40
41	SURF. FL. W. BED RM. 2ND FL.	66.7	61.9					66.2	66.7	41
42	SURF. CLG LIVING ROOM	66.3	66.1					66.8	67.2	42
43	SURF. FL. LIVING ROOM	67.2	64.0					67.0	66.0	43
	TIME GMD	6:57	10:51	7:57	12:00	7:54	11:55	8:28	12:36	

FUEL RECORD TEST HOUSE PROGRAM

READING BY H.W.J. DATE 2-23-42

TIME INDICATED IS EXACT TIME WHEN OIL IS ADDED TO WEIGHTING TANK	A AM	B	C	D AM
	9:30			10:30
WEIGHT INCREASED TO	304			375 7/8
WEIGHT BEFORE FILLING	201 1/2			238 3/8
WEIGHT OF OIL ADDED	102 1/2			137 1/8
WEIGHT LAST REG. READING	209			292 3/8
NEW "PREVIOUS WT" (TOP OF DELIVERY)	311 1/4			319 7/8

USE ONLY WHEN ADDING OIL TO SUPPLY TANK

READING BY H.W.J. READING BY F.B.N. DATE 2-22-42

TIME INDICATED IS EXACT TIME WHEN CLOCKS ARE READ AND WEIGHT OF OIL RECORDED	1 AM A	3 AM B	2 AM C	4 AM D
	6:30	7:40	7:05	8:15
	10:30	11:40	11:05	12:15
READINGS AT REG INTERVALS — PREVIOUS WEIGHT	209 5/8	311 1/2	304 3/8	304 3/8
	304 3/8	311 1/2	304 3/8	319 7/8
PRESENT WEIGHT	304 3/8	334 3/8	276 3/8	292 3/8
	334 3/8	349 3/8	294 3/8	373 3/8
CONSUMPTION (PERIOD BEFORE)	10 3/4	7 7/8	7 7/8	7 4 7/8
	9 3/4	7 7/8	7 7/8	7 4 7/8

TIME OF STARTS OF HEAT. PLANT OPERATION	HRS	MIN	HRS	MIN	HRS	MIN	HRS	MIN	HRS	MIN	HRS	MIN	HRS	MIN	HRS	MIN
CLOCK TIME																
HIGH + LOW FLAME	4	22	4	22	4	22	4	22	4	22	4	22	4	22	4	22
HIGH FLAME	1	48	1	48	0	38	0	38	0	38	0	38	0	38	0	38
LOW FLAME	2	34	2	34	3	20	3	33	3	20	3	33	3	20	3	43
PILOT FLAME	0	42	0	42	0	42	0	42	0	42	0	42	0	42	0	42

ELECT CURRENT CONSUMPTION KILOWATT HRS																
PRESENT READING	438.1		438.9		374.8		374.9		378.0		378.6		377.2		377.9	
PREVIOUS READING	433.4		438.1		372.7		374.5		377.6		378.0		376.7		377.2	
TOTAL CONSUMP.	4.7		0.8		2.1		0.4		0.4		0.6		0.5		0.7	
MONTH TO DATE	94.8		95.6		71.5		72.1		71.5		72.1		71.4		71.9	

INSIDE RELATIVE HUMIDITY																
TIME OF READING	6:56		10:55		8:01		12:05		7:28		11:28		8:37		12:40	
DRY BULB TEMP.	67.5		69.5		71.0		70.0		70.0		70.0		69.0		69.5	
WET BULB TEMP.	48.5		49.0		50.5		49.0		50.0		49.0		48.0		49.0	
% RELATIVE HUMIDITY	18.5		16.5		18.5		15.0		19.0		15.0		17.0		16.5	

STACK DATA READING BY T.D.K.																
TIME OF READING			8:25 AM				1:10 PM				10:10 AM				5:45 PM	
			HIGH	LOW	SMOKE		HIGH	LOW	SMOKE		HIGH	LOW	SMOKE		HIGH	LOW	SMOKE
ORSAT	CO_2		10.2	3.9			10.0	4.0	0.4		10.4	3.7			10.5	3.8	
	O_2		6.8	16.5			7.3	15.7	20.4		6.8	16.1			6.5	16.2	
	CO		0	0			0	0	0		0	0			0	0	
DRAFT (IN. OF WATER)			40	3.5			40	3.5	2.5		40	3.54			40	3.5	
STACK TEMP. DEG. F.			295	290			460	299	115		470	295			450	290	
GAL OIL CONSUMP. (PER HOUR)			.80	.20	.04		.80	.20	.04		.80	.20	.04		.80	.20	.04

SUNSHINE HOURS ENDING	6 AM	7	8	9	10	11 AM	12	1 PM	2	3	4	5	6	7
			0.1	0.2	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.87	

FIG. 3. EXAMPLE OF TEMPERATURE AND FUEL RECORD SHEETS

At certain periods during the heating season the bedrooms were ventilated. The inside lower sash was opened to a fixed height of 6 in. and a metal covered slot 1½ in. x 8 in., located in the bottom rail of the storm sash, was opened. Thus the amount of window ventilation was uniformly the same for all houses.

PROCEDURE

The tests were begun on October 10, 1941, and terminated on May 1, 1942. Thermostat settings were such that average air temperatures at the one-half level from floor to ceiling were as follows: first floor rooms, 70 F; second floor rooms, 68 F, except in uninsulated *House A* where, due to difficulty in maintaining a 68 F temperature, particularly during sub-zero weather, a

FIG. 4. TEMPERATURE READINGS BEING TAKEN WITH POTENTIOMETER AT CENTRAL PANEL

temperature 2 deg less, or 66 F, was established as an average value for the second floor rooms of this house. Inability to maintain the same temperature in the second floor of the uninsulated house as in those insulated can probably be attributed to greater duct loss, especially in the second floor runs and greater over-all heat loss through the uninsulated wall and ceiling construction. The first floor rooms of *House A* were, however, maintained at an average temperature of 70 F. The same thermostat settings at all houses were maintained for both day and night operation.

Air infiltration was controlled as far as practicable. No one except those actually taking readings was admitted to the houses during the test. Front and side doors were not used; each of the four houses was entered through the rear door, which opened directly into the kitchen. The doors from the kitchen to the dining room and to basement were kept closed at all times. The name of the person entering, the date, the time of entrance and exit, and the purpose of each visit were recorded. During two periods, January 18 to February 3, inclusive, and February 15 to February 25, inclusive, readings were taken at 4-hour intervals. The complete schedule is given in Fig. 6. During the balance of the tests, readings were taken at 12-hour intervals

beginning at 6:30 A.M. and 6:30 P.M. Example of record sheets on which the data obtained were recorded is shown in Fig. 3.

Each series of readings began at *House A* followed at 35-min. intervals by readings at *C, B* and *D*. The procedure at each was as outlined below:

1. Total time during which heating plant operated at stages of fire was determined from electric clocks.
2. Electric current consumption of heating plant was read from meters.
3. Fuel consumption was determined by weight (see Fig. 5).
4. Temperatures were read at central panel with portable potentiometer (see Fig. 4).
5. Wet- and dry-bulb temperatures, for determination of relative humidity, were obtained with hand-aspirated psychrometer.
6. During certain periods of the tests, bedrooms were ventilated at night. From October 10 to Feb 14, inclusive, two windows in each bedroom were opened to a fixed height as described previously. From February 15 to March 16, inclusive, no windows were opened. From March 17 to May 1, inclusive, both windows were opened. All changes in degree of ventilation were made at 6:30 p.m. and 6:30 a.m. When ventilation was provided, procedure during the 6:30 p.m. reading was as follows: Supply registers to both bedrooms were closed. Windows in both bedrooms were opened and the bedroom doors closed. At the 6:30 a.m. reading, this procedure was reversed—doors were opened, windows closed and registers opened. Changes in ventilation in all instances were made after temperature readings had been recorded.
7. Wind velocity was determined with electric cup anemometer.
8. Outside maximum and minimum temperatures were obtained by means of a maximum-minimum thermometer.
9. At frequent intervals the operation of the heating plant was thoroughly examined —this included analysis of the stack gases, measurement of stack temperature, draft, and check on flow of oil at each stage of fire. A system of switches was installed near the heating plant in the thermostat circuit to facilitate changing stage of fire during heating plant tests. The equipment required was Orsat apparatus, thermometer, draft gage and oil flow gage. The results of a typical test are shown in Fig. 3.

DISCUSSION OF RESULTS

Selection of Temperature Data for Analysis of Results

In a study of this kind covering as many as 1,000 temperature readings every 24 hours during a heating season, a vast number of temperature readings are accumulated. It is obvious that some method must be devised by which the information can be analyzed from a portion of the data, much in the way that public opinion is analyzed on the basis of a sample group.

Considerable judgment must be used in the selection of these sample periods, in order that each will be representative of a normal heating cycle. For example, (*a*) outside temperature at the beginning and end of period should be approximately the same; (*b*) trend of outside temperature change (increasing or decreasing) should be in the same direction at beginning and end of period; (*c*) individual outside temperature readings should be well distributed through a wide range of temperatures in order to obtain a wide distribution of points for plotting curves; (*d*) amount of sunshine during each period should be representative of an average condition that might prevail during the heating season; (*e*) amount of sunshine at beginning and end of period should be equal.

It is obvious that an excessive number of temperature readings obtained, for instance, only under conditions of decreasing outside temperatures or less than normal sunshine, should be avoided since temperatures within the

structure (inside surface temperatures, attic air temperatures, etc.) would be abnormally affected.

A group of the periods chosen for the detailed study is indicated in Fig. 6. Shown graphically, beginning at the bottom, are date of month, time, per cent of sunshine during each hour, schedule of readings, outside temperature, inside relative humidity based on outside absolute humidity, actual inside relative humidity and outside relative humidity at 6-hour intervals. Readings during these periods, as well as other periods not shown were used in preparing the curves in Figs. 8, 9, 10 and 11. With the exception of Fig. 11, which is plotted directly on the basis of actual temperature, all curves indicate

FIG. 5. OIL WEIGHING TANK, SCALE, AND CONNECTIONS

the temperature *difference* which existed between room air at one-half level and the particular surface or air temperature under consideration.

Inside Wall Surface Temperatures

Illustrated in the lower portion of Fig. 8 is the temperature difference between living room air at one-half level and the inside north wall surface (between studs). For instance, the top curve of this group indicates that in the uninsulated *House A* with — 10 F outside temperature, the wall surface temperature was approximately 15 deg less than the living room air. Of particular significance is the fact that the addition of insulation to the wall reduces the temperature difference between air and wall surface by nearly one-half under any outside temperature condition. For instance, at — 10 F outside, the temperature difference between air and wall was reduced to approximately 9 deg with the first thickness (0.9 in.) of insulation in *House B*. Each additional increase in thickness of insulation effected less than 1 deg change in surface temperature.

Solar radiation indicated its effect on inside wall surface temperature. As a result, the temperature difference between living room air and south wall between studs (not shown in curves) was approximately 1 deg less at any outside temperature than that indicated for north wall of *Houses B, C* and *D* in Fig. 8.

In the uninsulated *House A*, the temperature difference between air and south wall was approximately 2 deg less than that of the north wall. In either of the houses (with or without insulation), the south wall temperature approached the room air temperature at about 60 F outside temperature.

In comparing the temperature conditions of the south and north walls of the living room, the difference in construction should be considered. The south wall was of brick veneer, the north wall entirely frame. The *U* values of the north and south walls were approximately equal—however, there was

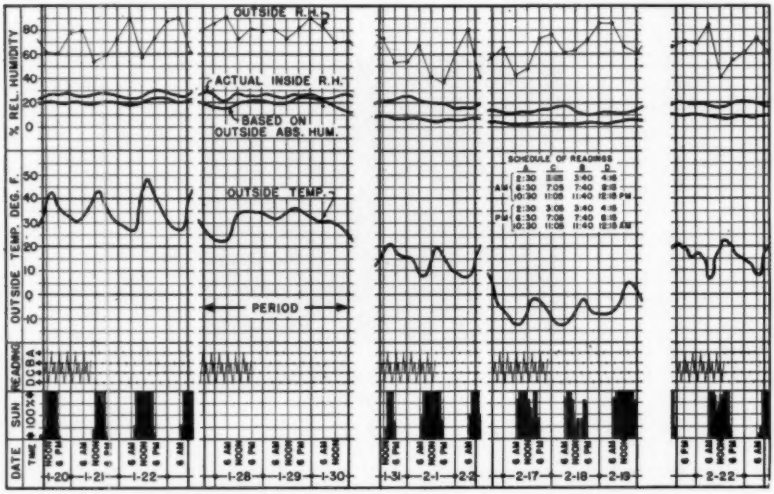

FIG. 6. TYPICAL PERIODS SELECTED FOR DETAILED ANALYSIS

probably some variation in time lag due to difference in heat capacity of the materials used.

Calculated wall surface temperatures, based on an inside surface conductance of 1.65 and outside surface conductance of 4.5 (10 mph wind), are indicated in this group of curves by broken lines. It will be noted that in each case a difference exists between the calculated and actual values, the variation being from 3 F to 4 F at — 10 F outside temperature and decreasing with rising outside temperature.

A similar variation between actual and theoretical values was found in a University of Minnesota study [1] and the results are indicated by points in Fig. 8. The writer concurs with the opinion of those investigators in that the inside surface conductance coefficients were probably much less than the commonly accepted 1.65 factor used in the calculations. This is believed to be due to the effect which the temperatures of other surfaces in the room

[1] Rowley, Algren and Lund, Engineering Experiment Station, Bulletin No. 18, University of Minnesota.

have in influencing the heat transfer to the wall by radiation. While only actual and calculated values for the uninsulated wall are shown in Fig. 8, the University of Minnesota study shows approximately the same variation in the case of insulated walls.

Assuming constant values for surface conductance and over-all wall transmission with changing outside temperatures, the calculated temperature difference shown in Fig. 8 should be represented theoretically by a straight line. The flattening of the actual curves at the lower outside temperatures probably can be explained by greater air movement over the inside surface of the wall resulting from increased blower operation at lower outside temperatures.

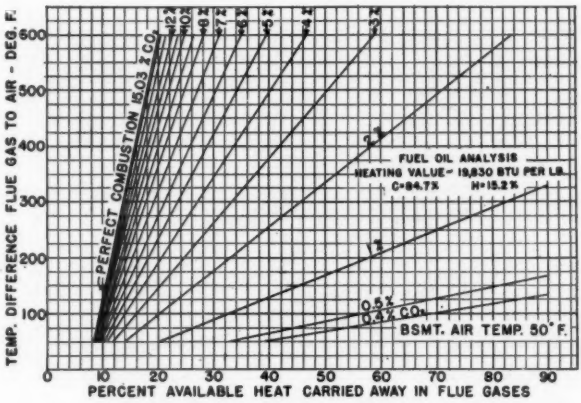
TEMP. DIFFERENCE FLUE GAS TO AIR - DEG. F.
PERCENT AVAILABLE HEAT CARRIED AWAY IN FLUE GASES
PERFECT COMBUSTION 15.03 % CO2
FUEL OIL ANALYSIS
HEATING VALUE = 19,830 BTU PER LB.
C=84.7% H=13.2%
BSMT. AIR TEMP. 50° F.
0.5%
0.3% CO2

FIG. 7. CHART FOR DETERMINING HEAT LOSS IN FLUE GASES

The effect which other surfaces have on the actual wall surface temperature in these tests would probably not be as pronounced if the room contained furniture, since the surface temperature of the furnishings would approximate that of the room air. However, variables affecting the surface conductance, such as character of surface (color and roughness) and the amount of air movement over the surface, make it difficult to predict accurately wall surface temperatures using standard surface conductance coefficients.

It is well established that the temperatures of surfaces enclosing a space affect the comfort of the occupant. If the temperatures of the surfaces are sufficiently low, the increased radiation of body heat to the surface may produce an uncomfortable condition. Studies at the ASHVE Research Laboratory, as reported in THE ASHVE GUIDE,[2] indicate that the dry-bulb temperature in the room tested must be elevated 0.3 deg to compensate for a reduction of 1 deg in the average temperature of three side walls. Thus, if the three exterior walls in the living room of each test house were uniformly of the same heat resistance, with no greater loss at studs or windows, and

[2] HEATING VENTILATING AIR CONDITIONING GUIDE, 1942, p. 59.

if the floor and ceiling surface temperatures were identical in all houses, then at — 10 F outside temperature we could assume that compared to *House B*, 1.8 deg (6 x 0.3 deg) higher air temperature would be required at *House A* for the same degree of comfort, to compensate for the 6 deg lower wall surface temperature at *House A*. See Fig. 8. Actually, however, in each of the test houses 17 per cent of the outside wall area of the living room is glass and at least 10 per cent studs, the surface temperatures of which we will assume are not affected by insulation. There remains a maximum of 73 per cent of the outside wall area, the temperature of which can be regulated by the addition of insulation. Therefore a change of 1 deg surface temperature

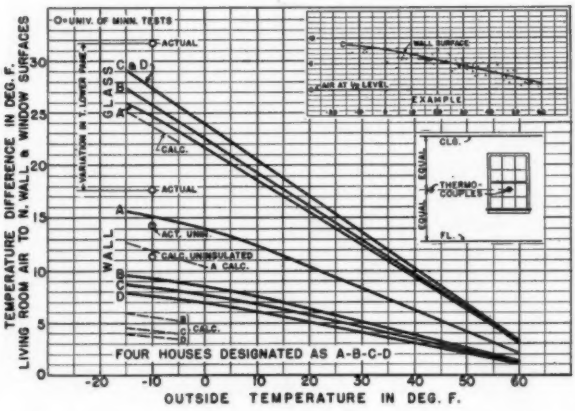

FIG. 8. TEMPERATURE DIFFERENCE—LIVING ROOM AIR TO NORTH WALL AND WINDOW SURFACES

between studs would actually be less than 1 deg in the over-all surface temperature of three walls, and would be offset by less than 0.3 deg change in air temperature. The same comparison between *Houses B* and *C*, and *C* and *D*, would indicate that the surface temperature difference of less than 1 deg, as measured between studs, would be compensated for by only a fraction of a degree change in air temperature.

These comparisons have been made at — 10 F, only for the purpose of discussion. At higher outside temperatures, changes in wall surface temperature as indicated for the walls shown in Fig. 8, would be even less pronounced.

Complete calculation of the mean radiant temperature, on the basis of temperature of *all* surfaces enclosing a room, would be necessary in order to establish the relative air temperatures which would be required in the four houses for the same degree of comfort. Some increase in temperature undoubtedly would be required in the uninsulated *House A* in order to give the same degree of comfort as in *Houses B, C* and *D*. This differential, of course,

would depend on the outside temperature. However, considering the temperature of every surface to which the occupant is exposed, including windows, partitions, ceiling, floor, furniture, wall hangings, etc., it is questionable if small variations in surface temperature between studs of outside wall areas of less than 1 deg, as indicated in comparison of *B* and *C*, and *C* and *D*, would affect noticeably the comfort of occupants, even though the same air temperature were maintained.

Wall surface temperatures directly over studs were analyzed. In the uninsulated *House A*, actual findings were compared with calculated values, deter-

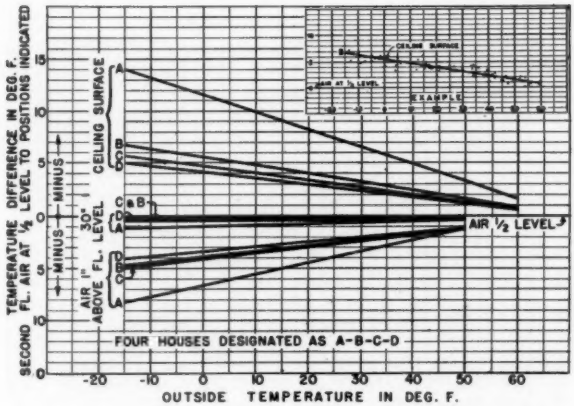

FIG. 9. TEMPERATURE DIFFERENCE—SECOND FLOOR AIR AT ½ LEVEL TO AIR AT 1 IN. AND 30 IN. LEVELS AND TO CEILING SURFACE

mined on the basis of parallel heat flow. Calculation indicated that at — 10 F outside temperature, the surface temperature over studs should be approximately 5½ deg higher than the surface between studs, due to insulating value of the stud itself. Actually the increase was between 1 deg and 2 deg. Probable explanation of this variation is that, due to high heat conductance of plaster board and plaster, heat flow occurred from the plaster directly over the stud into the colder adjacent areas. The tendency, therefore, was for the temperature over the stud to approach that of the area between the studs. The opposite condition was true in the insulated houses where the temperature over studs was 1 deg to 2 deg less than the temperature between studs, the greater difference occurring with increased thickness of insulation in *Houses C* and *D*.

Glass Surface Temperature

The temperature difference between living room air and glass surface of north windows is also shown in Fig. 8. It will be noted, for instance, that in *House A* the variation at — 10 F outside is 24½ deg and that this tempera-

ture difference increases at *Houses B, C* and *D*. Since surface conductance, even with double glass windows, comprises a large part of the entire resistance of the window structure, part of this variation may be attributed to decreased surface conductance due to less air circulation over the glass surfaces in the insulated *Houses B, C* and *D*. It is probably also reasonable to assume that part of the difference may be attributed to normal variations encountered in measurement of glass surface temperature where construction of the sash and enclosing frame, air infiltration, air movement, etc., are influencing factors. In the same University of Minnesota investigation [3] it was found, as indicated by points in Fig. 8, that the inside surface temperature

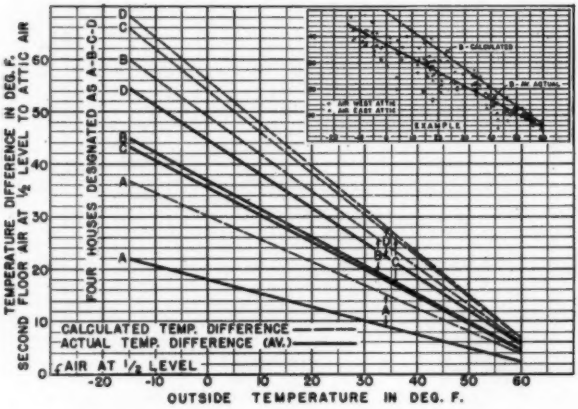

FIG. 10. ACTUAL AND CALCULATED TEMPERATURE DIFFERENCE—SECOND FLOOR AIR AT ½ LEVEL TO ATTIC AIR

of glass in the lower pane of a double-hung window (with storm sash), exposed to — 10 F outside temperature, varied 14 deg from top to bottom. In the test houses, the lower pane was divided into six parts with a muntin arrangement as shown in insert, Fig. 8. It is possible that the extending sill, jamb, and muntins further disturbed the air movement across the glass in such a manner as to contribute to the differences between measured values.

Second Floor Air and Ceiling Surface Temperatures

The difference in temperature between second floor air at one-half level and air 1 in. and 30 in. above the floor level and the ceiling surface, is shown in Fig. 9. The temperatures at 1 in. and 30 in. levels are important because the occupant's comfort depends largely on conditions in the lower portion of the room. For instance, in uninsulated *House A* at — 10 F outside, nearly 8 deg temperature difference existed between the air at one-half level and 1 in. above the floor. In the insulated *Houses B, C* and *D*, this temperature

[3] Loc. Cit., Note 1.

difference was 4 deg to 5 deg, or a reduction of approximately ½. Since the temperature 1 in. above the floor showed only a 1 deg variation between *Houses B, C* and *D*, it is evident that the different thicknesses of insulation as used in these tests, and applied to exposed walls and ceiling, have little effect on the air temperature at this level. Other studies made at *Houses A* and *D*, not shown in these curves, indicated that the temperature of the floor surface itself agreed very closely with that of the air 1 in. above the floor.

At — 10 F outside, the air at the 30-in. level in uninsulated *House A* was approximately 1 deg less than air at the one-half level. In the instance of insulated *Houses B, C* and *D*, however, the variation was not appreciable, being less than ½ deg with any thickness of insulation. The second floor ceiling surface in the uninsulated *House A* at — 10 F outside, was approxi-

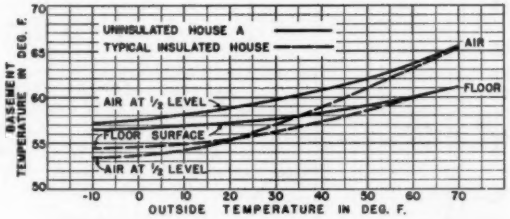

FIG. 11. BASEMENT AIR AND FLOOR SURFACE TEMPERATURES

mately 13 deg lower than the temperature of the air at one-half level. With the addition of insulation, this temperature difference was reduced by approximately ½.

Attic Air Temperatures

The temperature difference between air at one-half level of the second floor rooms and attic air is shown in Fig. 10. The solid lines indicate the average variation found by test with attic space vented with a louver in each gable having a net open area of 0.47 sq in. per square foot of flat ceiling area or 0.23 sq in. per square foot of total ceiling area, including the sloped portions. The broken lines indicate the variation as determined by calculation, using combined coefficients and assuming no attic ventilation. It is interesting to note that the average temperature drop through the ceiling construction, even with the attic vented, is less than that which would be indicated by a calculated attic temperature with no ventilation. This is consistent with the statement in the GUIDE that ordinary venting during the winter season may have only a minor effect on attic temperature.[4] It is assumed sometimes that venting the attic space would lower the attic temperature to that of the outside temperature. The findings here do not agree with that assumption. Higher attic temperature than indicated by calculation may be attributed to

[4] HEATING VENTILATING AIR CONDITIONING GUIDE, 1942, p. 112.

one or a combination of the following factors: (*a*) During the day, direct solar and scattered sky radiation even on cloudy days when a certain amount of radiation pierces the clouds. (*b*) During the night, some tendency for the construction, due to its heat capacity, to retain heat gained during the day. (*c*) Heat given up by the chimney.

Inconsistency as compared to calculated values also may be due to lack of exactness in methods of calculating attic air temperature, since the method

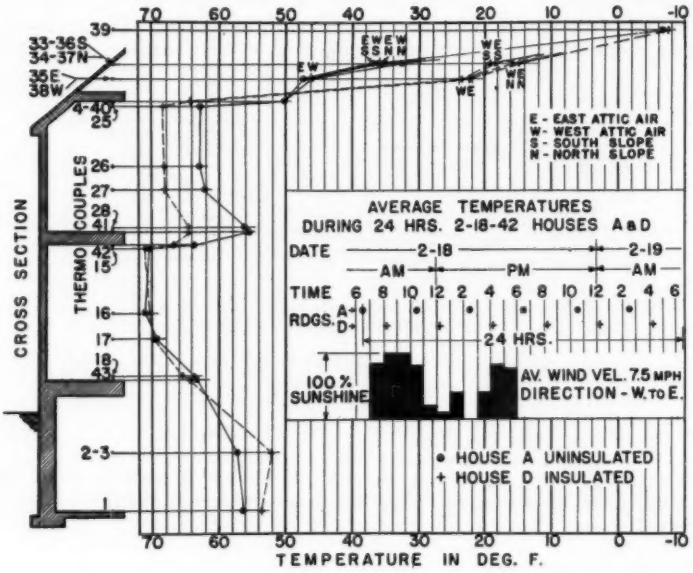

FIG. 12. TEMPERATURES AT VERTICAL SECTION THROUGH TWO HOUSES

is based on assumptions which may or may not be satisfied in individual cases.

In the calculation of fuel savings which can be effected with insulation, the portion of heat loss which is assumed to take place through ceiling and roof construction affects the over-all results appreciably. This variable, in combination with others which are encountered in the calculation of heat loss, is a contributing factor which results in many divergent claims made regarding the effectiveness of insulation. In many instances the calculated per cent savings is not in agreement with actual results. These tests indicate that further experimental work on heat loss through roof constructions involving varying amounts of ventilation, different types of roof construction, etc., would be desirable, in order that more accurate methods of calculating heat losses through this portion of the structure might be developed.

Basement Temperatures

Actual basement air and floor temperatures are shown in Fig. 11. Comparison is drawn between temperatures at uninsulated *House A* and a typical insulated house. The temperatures at *House A* are indicated by solid lines. Conditions in the insulated house are indicated by broken lines. The only heat supplied to the basement was that provided by normal loss from heating plant and ducts. Comparing air temperatures only, it will be noted that at — 10 F outside, the air temperature in the basement of the uninsulated house was approximately 4 deg higher than that in a typical insulated house. This can be explained by the fact that the heating plant in the uninsulated house operates more continuously and the resulting heat loss from the equipment is sufficient to increase the basement air temperature. Comparison of the floor surface temperatures at — 10 F indicates that while the temperature of the floor surface in the uninsulated house is less than the air temperature, the reverse condition is true in the insulated house. It would seem from these findings that the heat from the heating equipment in the insulated house is insufficient to supply the heat loss from the basement at these lower temperatures, and as a result the air temperature drops below that of the floor temperature. The ground temperature and normal lag between temperature of air and concrete floor, due to heat capacity of the latter, prevented the floor surface temperature from following the drop in air temperature during the relatively short periods of sub-zero temperature. As the outside temperature increased to 70 F the air and floor temperatures in both insulated and uninsulated houses approached 65 F and 61 F, respectively.

Temperature at Vertical Section in Houses

Indicated in Fig. 12 are average temperatures during a specific 24-hour period, plotted to show air and surface temperatures at the vertical section through *Houses A* and *D*. Reference to the vertical section at left indicates to approximate scale the relative position at which temperatures were measured, and the average temperature reading is indicated on the horizontal scale, reading from 70 F at left to — 10 F at right. The lines between points are merely connecting lines for easier reference to plotted points and are not intended to show the complete gradient through the structure. Each plotted point is the average of 6 readings, taken at 4-hour intervals during a 24-hour period. Of particular interest is (*a*) the lower floor and ceiling surface temperature in the first floor of uninsulated *House A;* (*b*) the temperature drop through the first floor ceiling, due to normal temperature gradient on first and second floors and the lower air temperature in the second floor rooms; (*c*) the small temperature drop through the uninsulated second floor ceiling construction and the corresponding large temperature drop through the insulated ceiling construction; (*d*) temperatures of the inside surface of the north and south slopes of the roof in both the east and west ends of the attic—the temperatures on the south slope being higher, due to solar effect; (*e*) the relatively high vented attic air temperature with reference to outside air in both insulated and uninsulated houses.

Heating Plant Efficiency

A typical flue gas analysis is recorded on data sheet in Fig. 3. In order to translate these results into efficiency of heating plant, the chart shown in

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Fig. 7 was prepared. The data for this chart were calculated on the basis of the ultimate analysis of the fuel oil used. Comparison of the results from each test made during the heating season with this chart indicated no appreciable variation in percentage of heat loss. From the results of the entire series of tests, certain average values for stack temperature and CO_2 were determined as follows: high fire, 470 F temperature with 10.4 per cent CO_2; low fire, 250 F temperature with 3.8 per cent CO_2. Reference to the efficiency chart indicates a stack loss of 19 per cent and 21 per cent respectively under these conditions of temperature and CO_2, with 50 F basement air. Radiation and unaccounted for losses must be added—from inspection these are assumed to be approximately 5 per cent at high fire and 3 per cent at low fire. Applying these corrections, the over-all heat loss becomes 24 per cent on either stage of fire. It is assumed, therefore, that for all practical purposes the over-all efficiency on both high and low stages of fire is 76 per cent. The calorific value of the fuel oil used was 19,830 Btu per pound; therefore on the basis of 76 per cent efficiency, the useful Btu per pound was 15,070.

Actual and Calculated Fuel Consumption

Shown in Fig. 13 is the actual accumulated fuel consumption during the heating season as plotted against the total degrees temperature difference per 24 hours. For any one day there exists as many degrees temperature difference as there are degrees Fahrenheit difference between the average outside temperature for the day and the average inside temperature. It was assumed, in compiling the information shown in Fig. 13, that the average temperature (first and second floors) maintained was 68 F in the uninsulated house and 69 F in insulated *Houses B, C* and *D*. The variance is due to a 2 deg lower average second floor temperature in *House A* than in *B, C* and *D*, with first floor in all houses 70 F. The points nearest the upper edge of the chart are located by plotting the total number of degrees temperature difference per 24 hours, accumulated from October 10, 1941, to May 1, 1942, against the total pounds of fuel consumed during the same period. The reference lines shown connect these particular points with the zero reference at the lower left without regard to the intermediate points. Based on this information, the average pounds of fuel oil consumed per degree temperature difference per 24 hours can be determined.

The fuel consumption during the period of test is indicated in Group I of Fig. 14. The consumption and savings in pounds and gallons of oil and the per cent savings accomplished with the various thicknesses of insulation are shown in table form. Of particular significance are the fuel savings; namely, 27.2 per cent for insulated *House B*, 31.3 per cent for *House C*, and 32.2 per cent for *House D*. The gallons of oil saved are 294.7, 338.6 and 347.8 respectively. The scale at the bottom of Fig. 14 indicates the heat loss in Btu per hour at -20 F outside temperature. The heat loss under conditions of actual test was determined on the basis of (1) pounds of fuel consumed per degree temperature difference per 24 hours as obtained from Fig. 13, (2) useful Btu per pound of fuel, and (3) difference between average inside air temperature and outside air temperature.

The actual fuel savings with temperature in second floor *House A* maintained at 68 F, are shown in Group II. The fuel savings for *Houses B, C*

and *D* are 29.2 per cent, 33.0 per cent and 34.0 per cent, respectively. The percentage savings in this instance are figured on the basis of the information taken from Fig. 13.

The calculated heat loss and savings in fuel, based on temperature information for basement space, attic, etc., as gained from this study, are shown in Group III. In this calculation, air infiltration is based on one volume change per hour, outside wind velocity 10 mph (assumed to be average for the Test House location), inside first floor temperature 70 F, second floor temperature 68 F at all houses, with an outside temperature of — 20 F. The

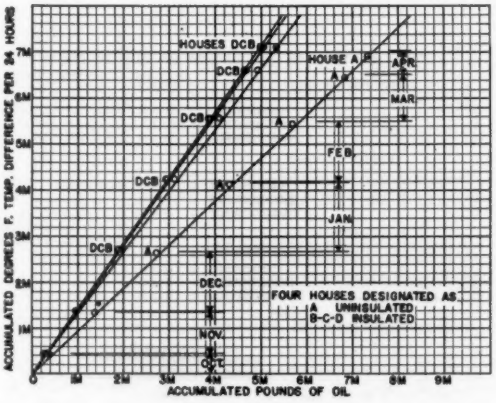

FIG. 13. FUEL CONSUMPTION *vs.* TEMPERATURE DIFFERENCE PER 24 HOURS

fuel savings for *Houses B, C* and *D* are 26.8 per cent, 33.5 per cent and 36.3 per cent respectively. The calculated heat loss and savings in fuel, assuming *outside temperature* in attic space and 32 F in basement, are shown in Group IV. In all other respects the basis for calculation was the same as for Group III.

In attempting to correlate calculated and actual heat losses, it is realized that there are many variables which effect the final results. In summarizing the information in Fig. 14, of particular significance, however, are the larger savings effected with the first increment of insulation and that subsequent thicknesses are relatively less effective. For instance, referring to Group II, the insulation in *House B* (0.9 in. thick) saves 29.2 per cent fuel during the heating season, the insulation in *House* C (1.7 in. thick) saves 33.0 per cent fuel, and the insulation in *House D* (2.3 in. thick) saves 34.0 per cent fuel.

The greatest spread between percentage savings with the least and greatest thickness of insulation used in these tests is actually 34.0 per cent — 29.2 per cent, or 4.8 per cent. Calculations based on test information indicated that this variation would be approximately 36.3 per cent — 26.8 per cent, or 9.5

per cent. A probable explanation for this discrepancy is that insulation used in *House B* is of a thickness which approximates the heat resistance of the framing members; therefore studs, etc., contribute effectively toward the reduction in overall heat loss. As the thickness of insulation increases, the loss through framing, plates, sills, sash weight boxes, diagonal bracing, fire stops, etc., becomes greater than the loss through the insulated portion of the stud space, and the effect of many of these factors in reduction of overall performance cannot be applied in calculations. The lower the theoretical coefficient the greater will be this percentage of error, due to greater heat loss through framing members and construction defects. Variations of this kind between theoretical and actual performance, as the insulation exceeds the insulating

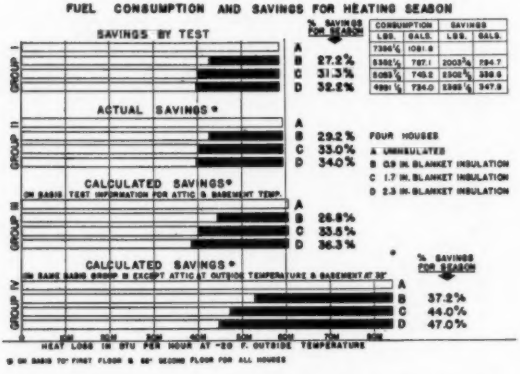

FIG. 14. ACTUAL AND CALCULATED FUEL CONSUMPTION AND SAVINGS FOR HEATING SEASON

value of the studs, have caused many engineers to arbitrarily adopt a minimum U value of 0.10.

The calculated heat loss and percentage fuel savings indicated in Group IV, Fig. 14, are shown merely to indicate the importance of proper use of attic temperature and basement heat loss information. All other factors used in calculating information for Group IV are identical with Group III except for the use of outside temperature in attic and 32 F in basement, rather than the temperatures as determined by this test.

CONCLUSIONS

Four test houses exposed to a wide range of weather conditions, were involved in this study. The construction of each was identical except as to the degree of insulation. *House A* was uninsulated; *Houses B, C* and *D* were insulated with blanket insulation in thicknesses of 0.9 in., 1.7 in., and 2.3 in. respectively. Following is a summary of the results obtained:

1. The installation of blanket insulation in *Houses B, C* and *D* effected a saving in fuel consumption by test of 27.2 per cent, 31.3 per cent and 32.2 per cent, respectively. With equal second floor temperatures in all houses, the savings become 29.2 per cent, 33.0 per cent and 34.0 per cent, respectively. Of particular significance is the relatively large savings in fuel with the first increment of insulation at *House B;* namely, 294.7 gal. Increasing the thickness of insulation in *Houses C* and *D,* increased the annual savings in fuel 43.9 and 9.2 gal, respectively.

2. The calculated heat loss based on information regarding temperature of attic space, basement, etc., as determined in this test, indicated reasonable agreement with the actual value. The savings as calculated, employing these refinements, were 26.8 per cent, 33.5 per cent and 36.3 per cent, respectively.

3. With attic temperature assumed to be the outside temperature and basement assumed to be 32 F, the calculated heat loss was increased approximately one-third and the calculated savings effected with insulation were 37.2 per cent, 44.0 per cent and 47.0 per cent, respectively.

4. At −10 F outside temperature, the inside surface temperature of north exposed uninsulated walls (between studs) was approximately 15 deg below that of the inside air. Insulation decreased the air to surface difference by approximately one-half. The variation in surface temperature between each of the thicknesses of insulation was less than 1 deg.

5. At −10 F outside temperature, the inside surface temperature of the second floor uninsulated ceiling was approximately 13 deg below that of the inside air at one-half level. The insulation decreased this temperature difference approximately one-half—the variation in surface temperature between each of the thicknesses of insulation being less than 1 deg.

6. At −10 F outside temperature, the inside surface temperature of glass in north windows with storm sash was approximately 24½ deg to 27½ deg below that of the inside air.

7. In the second floor rooms with −10 F outside temperature, the temperature difference between air 1 in. above the floor and one-half level was approximately 8 F without insulation. In the insulated houses this difference was reduced approximately one-half. The temperature change effected at this level with different thicknesses of insulation was not appreciable. The temperature difference between the one-half level and 30 in. above the floor was approximately 1 deg with no insulation and less than ½ deg with insulation.

8. The difference in temperature between air at one-half level in the second floor rooms and the average temperature of the vented attic, was less than the difference between second floor air and attic air calculated by combined coefficient method, assuming no attic ventilation.

9. At −10 F outside temperature, the basement air was approximately 57 F in the uninsulated house. In the insulated houses the air temperature was approximately 4 deg less, or 53 F. As the outside temperature increased, both of these temperatures approached 65 F. No heat other than normal loss from heating equipment was provided in the basement.

ACKNOWLEDGMENT

This study was sponsored by Wood Conversion Co. and the author wishes to acknowledge the valuable assistance and cooperation given by Professors F. B. Rowley and A. B. Algren of the University of Minnesota, both of whom have contributed liberally during the planning and execution of this work.

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DISCUSSION

N. D. ADAMS, Rochester, Minn.: The results reported in this paper are of great value. It would seem that the Society is in a position to inform the public from time to time concerning the proper materials and equipment which would permit the builder to make a wise economic choice. The current conditions are most confusing to the purchaser, who is confronted with a great number of viewpoints dealing with the selection of competitive materials, such as insulation.

The data reported in this paper together with other available Society data should be incorporated in a bulletin for the use of the consumer. Such a pamphlet issued by the Society might be entitled, The Economy of Good Building Construction, which could include such information as the following: (1) material—how much to use, cost and saving in fuel; (2) insulation—recommended thicknesses in walls, ceilings and wall exposure; (3) storm windows—weatherstripping; (4) heating system—reduction in size and cost due to the proper selection of materials and the annual fuel savings which can be obtained on a degree-day basis.

E. K. CAMPBELL, Kansas City, Mo.: I desire to compliment the authors of this paper on the very painstaking, tremendous amount of work involved in the preparation of this paper and results they are able to show.

Several years ago Mr. Close presented a paper dealing with the economical thickness limit of insulation, and pointing out that the limit was about an inch and a half thick. Beyond that the insulation did not pay its way. Up to that it very markedly did pay its way. That conclusion by Mr. Close is very closely borne out by this paper, which shows that two thicknesses very definitely paid their way but when the third thickness was added the increased saving was thought insufficient to justify the cost.

Another thing just pointed out by Mr. Adams in a conversation is that now when people come to us and want to know something about the saving in fuel resulting from proper insulation of the house, we are able to tell them pretty definitely; it is a valuable piece of work.

No. 1221

THE SPECIFIC HEAT OF THERMAL INSULATING MATERIALS

By Gordon B. Wilkes * and Carl O. Wood,** Cambridge, Mass.

Introduction

THE resistance to heat flow of the various thermal insulating materials under steady state conditions has been known for some time. In this case the conductivity value, k, and known conditions of temperature, thickness, and area are sufficient to determine the rate of heat flow through a particular material. However, in the unsteady state the specific heat and density are also important factors governing the performance of the materials, and in many cases they may be fully as important as the conductivity.

For a number of years the use of insulating firebrick of low heat capacity to replace regular firebrick in kilns and furnaces that are intermittently heated, has shown decisively that large savings in fuel and time can be obtained. McCullough [1] gives the results of tests on two furnaces, identical except for the type of lining, which were heated empty in order to compare the fuel requirements and heating time. The furnace with an insulating firebrick lining was heated to 2000 F with only one-third the quantity of gas required to heat the furnace lined with regular firebrick to the same temperature. Also the furnace lined with insulating brick reached 2000 F in approximately one third the time required for the other furnace.

In ovens and furnaces operated at lower temperatures (less than 1400 F) similar savings can be obtained. As an illustration, consider an oven insulated (1) with 4 in. of glass wool or (2) with 4 in. of 85 per cent magnesia. The oven, if maintained at a constant temperature of 500 F, will then lose very nearly the same amount of heat whether insulated as in (1) or (2). This statement is based on the premise that the k values are approximately equal for the mean temperature between 80 F and 500 F. Although this assumption is not true for all types of the two insulations, it will be satisfactory for this comparison.

If the period during which the oven is raised in temperature is considered, then the heat quantities involved in the two arrangements will be substantially

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* Professor of Heat Engineering, Massachusetts Institute of Technology. Member ASHVE.
** Assistant in Mechanical Engineering, Massachusetts Institute of Technology.

[1] Insulating Firebrick for Kiln Construction, by J. D. McCullough. (*Bulletin American Ceramic Society*, 20 [12] 1941.)

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, St. Paul, Minn., June, 1942.

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different. Also the time required to reach equilibrium will be considerably more in the second case than in the other.

Assume the oven insulated as in (1) to be heated from 80 F to 500 F. Taking the mean specific heat of glass wool to be 0.234 as determined in this investigation and assuming a density of 3 lb/ft³, then the heat required per cubic foot of material will be:

$$\frac{(500-80)}{2} \times 0.234 \times 3 = 148 \text{ Btu/ft}^3$$

Similarly the oven insulated with 85 per cent magnesia will require

$$\frac{(500-80)}{2} \times 0.284 \times 15 = 895 \text{ Btu/ft}^3$$

to heat the insulation.

Thus the heat requirements for the two insulations are in the ratio of 1 to 6. The heat capacity of the framework would reduce the ratio of total heat requirements according to the quantity of material in the framework and the type of construction. In no case, however, would this factor reverse the ratio since the same supporting structure was assumed in each case.

In comparing the times required to heat the apparatus to the desired temperature, the thermal conductivity must also be considered. The conductivity divided by the product of specific heat and density ($k/c\rho$) gives a term commonly referred to as *thermal diffusivity*. For materials of the same thickness, the heating time is inversely proportional to the diffusivity. For materials of the same diffusivity, the heating time is directly proportional to the square of the thickness. Consequently, for any material of any thickness, the controlling factor in heating time is

$$\frac{1}{k/c\rho} \times L^2 \text{ or } \frac{c\rho L^2}{k}$$

where L is the thickness.[2]

Since the two installations are the same thickness, however, the thermal diffusivity alone can be used for comparison of heating times.

Thus glass wool with a mean conductivity between 80 and 500 F of 0.44 Btu, hr^{-1}, ft^{-2}, F^{-1}, in. has a diffusivity of $0.44/0.234 \times 3 \times 12$ or 0.052 ft^2/hr. In order to have consistent units, the k factor of 0.44 is divided by 12 so that the new k is expressed in Btu, hr^{-1}, ft^{-2}, deg F^{-1}, ft.

The conductivity of 85 per cent magnesia can also be taken as 0.44. Then the diffusivity will be $0.44/0.284 \times 15 \times 12$, or 0.0086 ft^2/hr.

It is seen that the oven insulated with glass wool will reach 500 F in 0.0086/0.052 or approximately 1/6 of the time required for the magnesia installation.

These examples show that by consideration of heat capacity in addition to other properties a more satisfactory selection of insulation can often be made.

In order to provide data on the specific heat of thermal insulation an investigation has been carried out on some of the common materials. Little has been done along this line in the past. Until recently there has been very little interest among users of insulation either because they did not know the effect of heat capacity or they relied on past experiences. Furthermore, the

[2] *Heat Transmission*, by McAdams, Chapter II, 1933.

very nature of the materials presents considerable difficulty in measuring the specific heat.

The usual calorimetric apparatus employs a liquid, usually water, in which a heated specimen is dropped. The temperature rise of the liquid times its specific heat and weight will then give the total heat received from the specimen. The contact between liquid and specimen permits thermal equilibrium to be established in a comparatively few minutes. The heat gained from or lost to the surroundings during this short interval will then have only a small effect on the final accuracy.

However, because of the various binders and fireproofing chemicals present in insulating materials, a chemical reaction may take place when the substance is placed in water or other liquid. This of course would change the

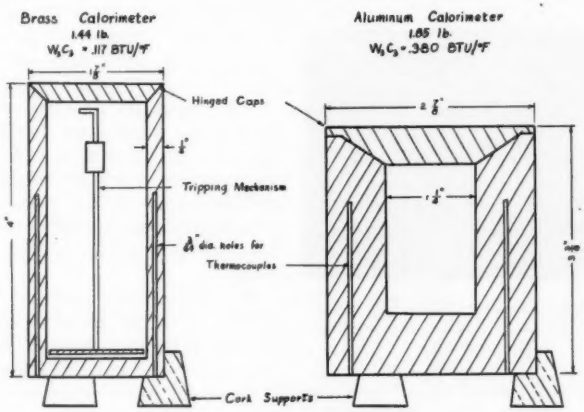

FIG. 1. DETAILS OF CALORIMETERS

measurement considerably. Also the fact that some of the materials will not sink, even though compressed to five or ten times their original densities, proved to be a considerable problem.

For these reasons it was found advantageous in this investigation to use a dry calorimeter [3] in which a block of metal was substituted for the liquid. The materials were compressed to pellets with five or ten times the original densities but the thermal conductivity was still relatively low. This, along with the high thermal resistance between the sample and calorimeter, hindered heat flow to a considerable extent.

In order to hold the time required to reach equilibrium to a minimum, it was necessary to use as small a sample as possible. At the same time the sample had to be large enough to maintain satisfactory accuracy in weighing. Furthermore, as it was found necessary to cover the sample with metal foil

[3] Improved Metal Calorimeter for Determination of Specific Heats of Metals, Oxides and Slags, by Grosse and Dinkler. (*Stahl und Eisen*, 47 [11] 1927.)
The Specific Heat of Magnesium and Aluminum Oxides at High Temperatures, by G. B. Wilkes. (*Journal American Ceramics Society*, 15 [1] 1932.)

to prevent loss of material, the sample had to be large enough so that the foil heat capacity would be relatively unimportant.

Because of these limitations, the time required for the sample to reach equilibrium with the calorimeter was fixed at approximately 50 min. Although this comparatively long time is not usually satisfactory for calorimetric measurements, its effect on the final accuracy was reduced considerably by calibrating the calorimeter under practically the same conditions as in an actual determination.

APPARATUS AND METHOD

The mean specific heat between 80 F and temperatures ranging from 212 F to 1350 F were determined experimentally for 15 materials. In addition, the variation of true specific heat with temperature was calculated for those materials which, by virtue of their use in industry, seemed to warrant determinations at more than one temperature.

FIG. 2. THE APPARATUS FOR DETERMINING THE SPECIFIC HEAT OF THERMAL INSULATION

The apparatus consisted essentially of a metal calorimeter complete with an *adiabatic* jacket, a potentiometer and thermocouples for measuring the temperatures of the specimen and calorimeter, and furnaces for heating the specimen to the desired temperature.

The calorimeter itself was a cylindrical block of metal with a hole drilled part way through parallel to the axis. The open end was covered by means of a hinged cap. Three small corks glued on the other end served as supports (see Figs. 1 and 2).

The temperature of the calorimeter was measured with a double copper-constantan thermocouple and a portable precision type potentiometer. The two measuring junctions of the thermocouple were placed inside small glass tubes and inserted in holes drilled into the calorimeter wall from the bottom. The reference junctions were kept in a thermos bottle filled with cracked ice and water.

The jacket consisted of two spun copper cans of unequal size, placed one inside the other so as to have a ¾-in. space between at all points except the top. The space was filled with water. The two cans were then connected across a 110 volt power line with a switch in the circuit so that the jacket could be heated as desired. In order to increase the efficiency of the jacket, the outer can was surrounded by 2 in. of insulation in a still larger container.

The temperature difference between jacket and calorimeter was indicated by means of a differential thermocouple and a spotlight galvanometer. One junction of the couple was placed inside a thin glass protection tube and immersed in the jacket water. The other was soldered to the outer surface of the calorimeter.

In determining the specific heat of a given material, a sample was first prepared by compressing it under a pressure of about 100,000 lb per square

TABLE 1—RESULTS OF ALL MATERIALS TESTED

MATERIAL	MEAN SPECIFIC HEAT[a]	TEMPERATURES DEG FAHR.[b]		
		t_2	t_3	MEAN
Lead slag wool	0.178	88	212	150
	0.198	88	484	286
	0.235	86	1044	565
	0.235	86	1350	718
Glass wool (with binder)	0.196	87	212	150
	0.231	86	490	288
Rock wool	0.201	86	212	149
	0.212	87	455	271
	0.250	87	1063	575
	0.250	85	1219	652
High temperature block	0.203	86	212	149
	0.227	90	490	290
	0.263	89	1049	569
	0.269	84	1336	710
Silica Aerogel	0.205	82	212	147
	0.222	87	455	271
	0.267	86	1056	571
	0.274	83	1177	630
Vermiculite	0.205	85	212	149
	0.236	90	490	290
Glass wool	0.210	88	212	150
	0.233	83	487	285
	0.275	87	1051	569
	0.279	87	1161	624
Corrugated asbestos paper	0.245	82	212	147
Rock wool with asphaltic binder	0.247	85	212	149
85 per cent Magnesia	0.276	88	212	150
	0.283	88	470	279
Wood fiber blanket	0.330	85	212	150
Hair felt	0.334	84	212	148
Wood fiber board	0.341	83	212	148
Expanding paper blanket	0.349	84	212	148
Cork board	0.417	87	212	150

[a] Average of several runs where more than one was made. Corresponding temperatures are also averages.
[b] Temperatures given to nearest degree.

inch so as to form a solid pellet. This pellet was then cut down to the desired weight (from 2 to 15 g (grams) depending on the material and the temperature to which it was to be heated, correlated with a calorimeter tem-

perature rise of from 4 to 7 F). The sample was then covered with metal foil to prevent loss of material and reweighed to determine the amount of foil. Aluminum foil was used for the low temperature runs and platinum foil for temperatures above 1100 F.

The sample was heated and maintained at a known temperature for several hours. It was then dropped into the calorimeter and the temperature rise of the latter measured.

The jacket was kept at very nearly the same temperature as the calorimeter at all times in order to reduce heat losses from the calorimeter. The average

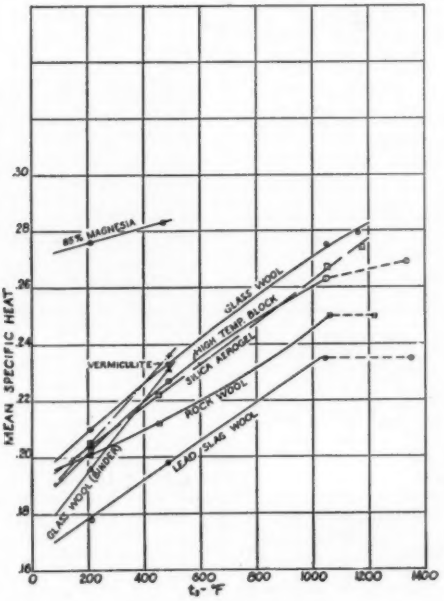

FIG. 3. MEAN SPECIFIC HEAT BETWEEN T_3 AND 80 F *vs.* T_3

temperature difference during a run on any of the materials was only 0.05 F. Heat exchange between the two parts was further reduced by covering both the calorimeter and the inside surface of the jacket with aluminum foil.

Due to the limitations on the size of the samples, it was found necessary to use two calorimeters; one for the low temperature runs (212 F-500 F) and the other for the higher temperatures. The heat capacities of the two calorimeters were 0.117 Btu/F and 0.380 Btu/F, the smaller being made of brass, the other of aluminum.

The calorimeters were calibrated with a very small electric heater. After placing the heater inside the calorimeter and making sure that the tempera-

ture of the apparatus stayed constant, current was supplied for a period of twenty minutes, after which the calorimeter was allowed to reach equilibrium. Current and potential readings taken at one minute intervals gave the power input to the heater. The heat capacity of the calorimeter could then be calculated from the relation:

$$E\,I^{\tau} \times 3.413 = W_3\,C_3\,(t_2 - t_1)$$

where: E = voltage drop across heater, volts
I = amperes through heater,
τ = time heater was on, hours,
W_3C_3 = heat capacity of calorimeter (W_3 = weight)
C_3 = specific heat, Btu/F
t_1 = initial temperature of calorimeter, F
t_2 = final temperature of calorimeter, F

Corrections for the heat capacity of the heater itself were negligible, amounting to only 0.0002 Btu/F as calculated with specific heat and weight data.

The samples were heated in three different furnaces depending on the temperature desired. A small steam jacket with radiation shield was used for heating the sample to 212 F, the sample being held in a pyrex glass test tube which hung in the steam bath. In transferring the sample from the steam bath to the calorimeter, the test tube was lifted over to the calorimeter and inverted, care being taken that no water droplets fell into the calorimeter. Due to the relatively large heat capacity of the test tube, heat losses from the sample were negligible during the transfer period. The sample was exposed to the room for only about two seconds as it dropped into the calorimeter. The steam temperature was calculated from the barometric pressure.

An electrically heated furnace was constructed for the 500 F determinations. This was small enough so that it could be lifted directly over the calorimeter for dumping the sample.

For temperatures above 500 F a larger electrically heated furnace was needed. Here again a container of relatively large heat capacity held the sample while in the furnace and during the transfer to the calorimeter. A porcelain tube sealed at one end was used in this case.

TABLE 2—CONSTANTS FOR THE TRUE SPECIFIC HEAT EQUATION

$$C_T = A + Bt + Ct^2$$

MATERIAL	A	B	C	TEMP. RANGE DEG FAHR.
Lead Slag Wool	0.159	0.000136	0	80 to 1050
Rock Wool	0.189	0.0000738	0.0000000450	80 to 1050
Silica Aerogel	0.184	0.000142	0	80 to 1180
High temperature block	0.174	0.000202	−0.0000000611	80 to 1050
Glass wool	0.182	0.000192	−0.0000000450	80 to 1170
Glass wool with binder	0.159	0.000253	0	80 to 500
Vermiculite	0.171	0.000226	0	80 to 500
85 per cent Magnesia	0.268	0.0000533	0	80 to 470

In all determinations above 212 F, the temperature of the sample in the furnace was measured with a thermocouple which laid alongside the sample.

The mean specific heat was calculated from the experimental data as follows:

$$\frac{W_1}{454}(t_3 - t_2)\,C_1 + \frac{W_2}{454}(t_3 - t_2)\,C_2 = W_3\,C_3(t_2 - t_1)$$

where: W_1 = weight of sample, grams
W_2 = weight of foil, grams
t_1 = initial temperature of calorimeter, Fahrenheit
t_2 = final temperature of calorimeter, Fahrenheit
t_3 = temperature of sample in furnace, Fahrenheit
C_1 = mean specific heat of sample between t_3 and t_2
C_2 = mean specific heat of foil between t_3 and t_2
$W_3\,C_3$ = heat capacity of calorimeter, Btu/Fahrenheit

Then solving for C_1:

$$C_1 = \frac{W_3\,C_3(t_2 - t_1) - \frac{W_2}{454}(t_3 - t_2)\,C_2}{\frac{W_1}{454}(t_3 - t_2)}$$

The results for all the materials tested are given in Table 1. Fig. 3 shows these data plotted as mean specific heat *vs.* t_3.

In order to find the variation of true specific heat, C_T, with temperature, the following relation was assumed:

$$C_T = A + Bt + Ct^2$$

where A, B, and C are constants. Considering the heat involved in raising one pound of material from t_2 to t_3:

$$C_1(t_3 - t_2) = \int_{t_2}^{t_3} (A + Bt + Ct^2)\,dt$$

Or carrying out the integration gives:

$$C_1(t_3 - t_2) = \left[At + \frac{Bt^2}{2} + \frac{Ct^3}{3} \right]_{t_2}^{t_3}$$

If determinations of C_1 are made for three different temperature differences then appropriate values of t_2, t_3 and C_1 can be substituted in the equation to obtain three simultaneous relations and the constants A, B, and C determined.

In several cases the relation between C_1 and $(t_3\text{-}t_2)$ proved to be a straight line so that only two equations were needed.

The constants found in this way are listed in Table 2. Fig. 4 shows true specific heat plotted against temperature.

Discussion of Results

Referring to Fig. 3 it is seen that above 1050 F the mean specific heat of the high temperature block, rock wool and lead slag wool change markedly. This is due to a chemical change which is further evidenced by a decided

change in color. This variation in specific heat is not important, however, since only a very small layer of the insulation would actually reach a temperature above 1050 F.

Since the performance of materials in unsteady heat flow conditions involves also the density and conductivity, a fair comparison is not obtained from specific heat data alone. As the density of most thermal insulating materials

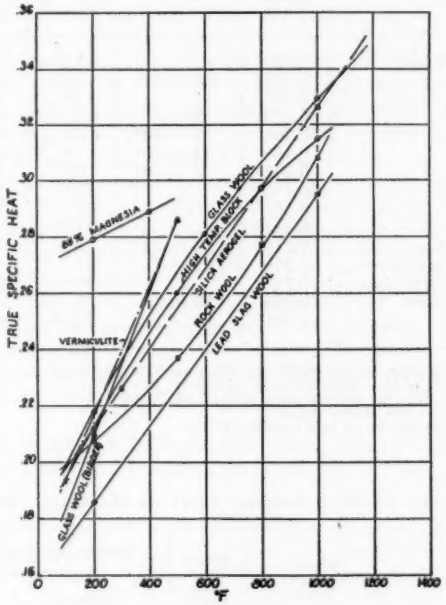

FIG. 4. TRUE SPECIFIC HEAT *vs.* TEMPERATURE

is a highly variable quantity, it would be impossible to rigidly compare their heat capacities and thermal diffusivities.

However, as a matter of interest the mean heat capacity values between 80 and 212 F are listed in Table 3 in the order of increasing value. Also the corresponding thermal diffusivities are given. The densities and conductivities used were obtained from manufacturers' circulars where possible. Otherwise values were taken from tests conducted at the Massachusetts Institute of Technology or from the HEATING VENTILATING AIR CONDITIONING GUIDE, 1942.

Assuming the same thickness of material in all cases, Table 3 shows that the expanding paper blanket insulation will not only require the least amount of heat to raise its temperature from 80 to 212 F but also will heat up faster

TABLE 3—SPECIFIC HEAT, HEAT CAPACITY, AND THERMAL DIFFUSIVITY OF VARIOUS INSULATING MATERIALS

NOTE: *Density values given are only approximate. This table is intended for a rough comparison and not for a rigorous statement of properties.*

	MEAN VALUES BETWEEN 80 F AND 212 F				
MATERIAL	SPECIFIC HEAT	DENSITY[a] VARIATION	AVERAGE DENSITY[a]	HEAT CAPACITY[b]	THERMAL DIFFUSIVITY[c]
Expanding paper blanket	0.349	...	1.6	0.55	0.046
Glass wool	0.210	1.5 — 4.5	3	0.63	0.036
Silica Aerogel	0.205	...	4*	0.82	0.020
Wood fiber blanket	0.330	2 — 4	3	0.99	0.023
Rock Wool	0.201	2 — 10*	6*	1.2	0.019
Lead slag wool	0.178	4 — 10*	7*	1.3	0.017
Vermiculite	0.205	8 — 10*	9*	1.8	0.025
Glass wool (binder)	0.196	...	12*	2.3	0.011
Cork board	0.417	...	8	3.3	0.0068
Hair felt	0.334	...	11**	3.7	0.0061
85 per cent Magnesia	0.276	12 — 18	15	4.1	0.0086
Corrugated asbestos paper	0.245	12 — 22	17	4.2	0.0089
Rock wool with asphaltic binder	0.247	...	17	4.2	0.0066
High temperature block	0.203	18 — 28	23	4.7	0.011
Wood fiber board	0.341	12 — 17*	14.5*	5.0	0.0055

[a] lb/ft³. Values are as given by manufacturers, unless specified as follows:
* From tests conducted in Heat Measurements Laboratory, M.I.T.
** HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1942, p. 95.
[b] Specific heat times average density Btu/ft³/F.
[c] Thermal conductivity divided by heat capacity, Ft²/hr.

TABLE 4—MEAN SPECIFIC HEAT OF MATERIALS TESTED

MATERIALS	MEAN SPECIFIC HEAT	MEAN TEMP. DEG FAHR.	RESULTS OF THIS INVESTIGATION	
			Materials	Mean Specific Heat
Slag wool[a]	0.17	77	Lead slag wool	0.17
Baked cork[a] (slab)	0.43	77	Cork board	0.417[f]
Granulated cork[a]	0.44[e]	77	Cork board	0.417[f]
Cork (expanded but not impregnated)[b]	0.433	91	Cork board	0.417[f]
Natural cork[b]	0.419	124	Cork board	0.417[f]
Kapok fiber[b]	0.320	65		
Cellulose[e]	0.319—0.327	*	Organic insulators	0.330—0.417[f]
Asbestos[e]	0.25	*	Corrugated asbestos paper	0.245[f]
Glass[d]	0.14—0.23	*	Glass wool	0.210[f]

[a] Heat Insulators in *Food Investigation*, Special Report No. 35, National Physical Laboratory, London (1929).
[b] Specific Heat of Building and Insulating Materials, by M. Kinoshita. (*Journal Society of Chemical Industries*, London, 36, 1917, 597).
[c] International Critical Tables, Vol. II, 237 (1927).
[d] *Ibid.*, Vol. II, 93.
[e] Material contained 3.4 per cent moisture.
[f] Mean temperature 150 F.
* Temperatures not given.

than any of the materials tested. Of the inorganic insulators tested, glass wool has the lowest heat capacity and the highest diffusivity.

Of course in many instances a high heat capacity and low diffusivity is desired. An annealing furnace, for instance, might require slow cooling if material is left in it after being heated to the desired temperature. In this case 85 per cent magnesia or high temperature block insulation might be the best choice.

Allcut [4] notes several instances in Canada where the use of thick insulation to obtain a large heat capacity has been very satisfactory in maintaining constant temperatures and low fuel costs.

He cites the example of a hospital in Prince Edward Island which utilized insulation 10 in. thick and required less than half the heating capacity of similar hospitals with ordinary construction. This saving was attributed to the effect of time lag in alternately storing and returning heat from both the heating system and solar radiation.

An instance where several unheated packing sheds and storage buildings utilized the effect of heat capacity was also mentioned by Professor Allcut. Because of the large heat capacity of the walls, the buildings have been consistently maintained at temperatures above 30 F even though outdoor temperatures often fall below zero.

Comparison of the specific heat values with those of other observers shows good agreement. This correlation, however, was possible for only a few of the materials because of the meager data in the literature.

Since heat insulators are seldom pure substances, the small variations in values of different observers are likely due mainly to slight differences in composition.

In Table 4 are listed values obtained from different sources along with those obtained on similar materials during the course of this investigation.

PRECISION

The precision obtained was ± 1 per cent as derived by the following considerations:

The temperature rise of the calorimeter was measured with a double copper-constantan thermocouple. Thus, since the potentiometer could be read to ±0.001 millivolts, the actual value was known to ±0.0005 millivolts. Therefore, the third decimal place was fixed with sufficient accuracy, the temperature rise being greater than 0.100 mv. (single couple).

The temperature of a sample in the furnace was measured with a single copper-constantan thermocouple at the lower temperatures, and a single chromel-alumel couple above 1000 F. Calibrations for both couples were known to well within ± 1 per cent error. Since the couple lay alongside the sample and touching it inside the heater, and since sufficient time was given for the whole to come to equilibrium, there is little chance for error in the assumption of a ± 1 per cent accuracy for this measurement.

The temperature of the steam bath was known to well within ± 1 per cent.

The time required to transfer a sample from furnace to calorimeter was only two or three seconds. In this short interval, the quantity of heat loss was negligible. In all runs the sample was held in a container of relatively large heat capacity, while in the furnace and until dropped into the calorimeter. The distance through which the sample fell was only about six inches.

[4] Properties of Heat Insulating Materials, by E. A. Allcut. (*The Engineering Journal*, 24 11 1941.)

The heat losses from the calorimeter to the surroundings were reduced to a minimum by means of the *adiabatic* jacket and aluminum foil as explained previously. The effect of these losses was further reduced by the calibration procedure.

Check runs were made on all but six determinations in which cases the values fell on the curve of mean specific heat vs. t_3

The weight of the samples was measured to $\pm$ 1 per cent.

The heat capacity of the foil amounted to no more than 4 per cent of that of the sample plus foil. The value of that quantity could therefore be in error by as much as 25 per cent without affecting the results.

ACKNOWLEDGMENT

The authors are indebted to L. R. Viäney for his help and suggestions on the construction of the apparatus.

DISCUSSION

JOHN JAMES, New York, N. Y.: Prior to the reporting of these tests, only meager data have been available in the various engineering handbooks on the specific heats of common insulating materials. Hence, the results reported in this paper will serve a useful purpose in the literature of the Society. At the present time some analytical work is being conducted at the ASHVE Research Laboratory on the heat flow through various combinations of wall construction, and as one phase of the calculations involves heat capacity, it will be readily observed that the data in this paper will be of value. The Society is now sponsoring a cooperative research project at Columbia University to determine the heat gain through walls utilizing an electrical analyzer. The accuracy of the results obtained from this electric device depends largely on the reliability of constants fed into the machine. One of these constants is the specific heat of a material and for that reason the results given in this paper will be of great interest to those working on this project.

E. C. LLOYD, Lancaster, Pa.: The authors are to be congratulated on an excellent paper which supplies some much needed data on the specific heat of insulating materials commonly used in both high temperature and low temperature work. Heretofore, values for the specific heat of many of these materials have been lacking, and the paper therefore serves a long felt need by providing reliable information on the 15 materials tested.

An inspection of the results given in Table 1, places the materials in three definite classes—inorganic materials showing a specific heat of 0.2 to 0.25, organic materials other than corkboard showing a specific heat of 0.33 to 0.35, and corkboard showing a value of 0.4. Since corkboard, as well as two of the organic materials tested, is largely cellulose, it is difficult for me to see why this difference should come and I should like to inquire of the authors if any tests were made of wood as I feel that possibly the woody materials would occupy a place between the fibrous materials and that occupied by cork, or might possibly show a specific heat even in excess of the value given for corkboard.

PROFESSOR WILKES: In answering Mr. Lloyd's comments, I regret to state that no determinations of the specific heat of wood were made during this investigation. According to *Marks' Handbook*, the specific heats of pine, oak and fir are between 0.45 and 0.67, although the source of these data is not given. The specific heat of cellulose is given as 0.32 by the International Critical Tables. The wood fiber blanket and the wood fiber board have specific heats very close to this cellulose value. I presume that corkboard has a higher specific heat than pure cellulose on account of the resins, etc., that remain in the material and which probably have a much higher specific heat than cellulose. The value for corkboard is not as high, however, as the values for three types of wood as given by Marks.

XUM

No. 1222

SIMULTANEOUS HEAT AND VAPOR TRANSFER CHARACTERISTICS OF AN INSULATING MATERIAL

By F. G. Hechler,* E. R. McLaughlin,** and E. R. Queer,† State College, Pa.

THE primary purpose of a building insulating material is to restrict the flow of heat from one space to another. Any action which tends to thwart this purpose may be a vital consideration. Free moisture within a material nearly always decreases its thermal resistance and hence the water vapor transfer characteristics of the material are of interest.

There are two major forces which tend to transfer vapor through building insulation. The first of these which affects all materials is the differential that exists between the vapor pressure on opposite sides of the insulation. This pressure differential causes a net diffusion of vapor in a manner postulated as Fick's Law, which in the integrated mathematical form is:

$$W = DAt\frac{dP}{dx}$$

This states that for a given area A the weight of vapor W which diffuses through the material is proportional to the time t and the vapor pressure gradient $\frac{dP}{dx}$.

Organic materials also transfer moisture due to another force known as hygroscopic action. These materials assume a moisture content which is proportional to the relative humidity of the air. This absorbed moisture tends to migrate from spaces of higher to spaces of lower moisture concentration, and the effect is to transfer vapor from a space of higher to a space of lower relative humidity.

These two forces may operate in the same direction or they may be opposed and the vapor transferred through a material is the net result of the two forces acting simultaneously. The moisture content of the material is also the net result of the two types of moisture movement.

Test equipment which provides for these dynamic conditions has been constructed as a guarded hot box with facilities for determining the moisture

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Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, St. Paul, Minn., June, 1942.

XUM

and vapor conditions. A cutaway view of the box and the test room is shown in Fig. 1. The specimen *S* is clamped to the guard box to form the sixth side (top, in sketch) of both the test box and the guard box. The small test box inside, 18 x 18 x 15 in., fits snugly against the specimen so that it is sealed to both air and moisture along the edges by means of a rubber gasket. The inner surfaces of this box are one copper sheet shaped to form. The same is true for the outside surfaces. These copper sheets are thermally and electrically insulated from one another. The larger outside box supports this small test box and provides a 4-in. guard space around five sides of it.

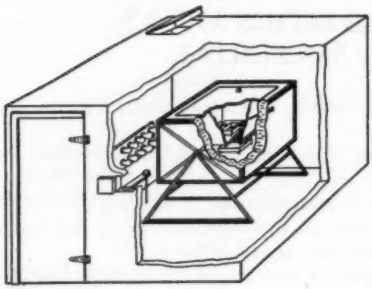

FIG. 1. (*above*) CUTAWAY VIEW OF GUARDED HOT BOX

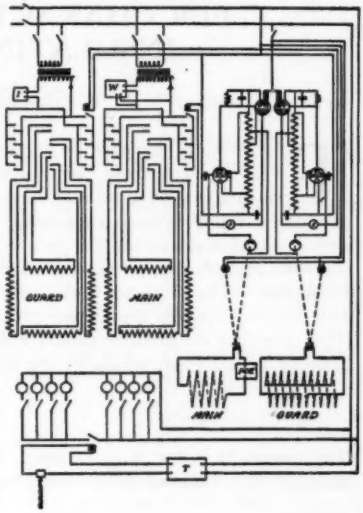

FIG. 2. (*right*) POWER AND CONTROL CIRCUITS

The inside surface of the guard box is lined with copper to provide a moisture seal. All copper surfaces are painted with shellac and lampblack. The guard box is built of cork insulation and ⅜-in. plywood braced with angle iron. The entire box is mounted so that it may be rotated to give the specimen any angular position with the horizontal, heat flowing up or down through the specimen. The entire box and its mounting is housed in a small room in which the temperature is controlled by means of a thermostat, shown in the foreground of Fig. 1. Heat is supplied by the lampbank along the left wall and all heaters are shielded so that they do not radiate directly onto the specimen or any of the thermocouples. Vents in the top and walls of the test room allow cold air to circulate through the small room from a larger cold room maintained at 0 F.

The power circuit for the entire set-up is shown in Fig. 2. The heaters in the test box are connected in series with a watt-hour meter *W* and a group of switches so that any heater may be put into either continuous or intermittent service under the control of the thermostat. The relay in this circuit

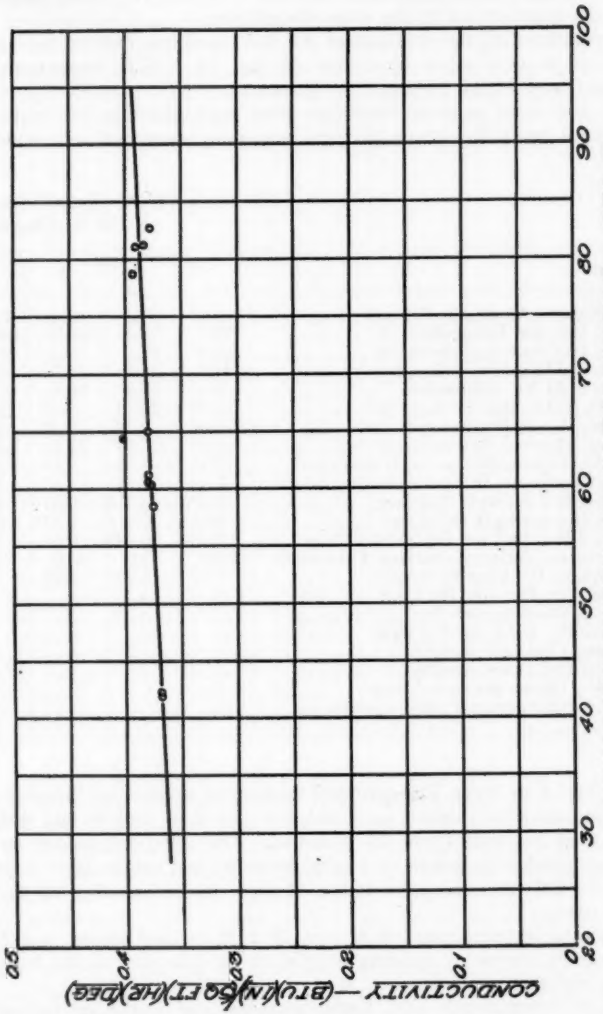

FIG. 3. RELATIONSHIP OF CONDUCTIVITY TO MEAN TEMPERATURE FOR 0.795-IN. WOOD FIBER BOARD

is actuated by the amplifier [1] in response to a photo electric tube. This gives continuous automatic air temperature control. A similar circuit maintains the air in the guard space at the same temperature.

The temperatures of the surfaces of the test specimen and of the air on both sides of it were taken by means of No. 30 A.W.G. copper-advance thermocouples and a high precision potentiometer.

Constant and equal relative humidities were maintained in the main test box and guard space by means of pans containing saturated salt solutions.

TABLE 1—STEADY STATE TESTS—
(0.795 in. Fiber board,

NUMBER OF TEST	1	2	3	4
Mean Temperature, Air to Air, Fahrenheit	39.7	59.2	63.0	77.1
Temperature, Hot Air, Fahrenheit	70.3	92.4	106.0	109.0
Temperature, Hot Surface, Fahrenheit	61.6	82.0	93.4	99.4
Temperature, Cold Surface, Fahrenheit	21.9	38.9	36.5	58.0
Temperature, Cold Air, Fahrenheit	9.1	25.9	19.9	45.1
Heat Transfer, Total, Btu per sq ft (hr)	18.37	20.61	27.23	20.44
Heat Transfer, Vapor, Btu per sq ft (hr)	0.00	0.00	0.00	0.00
Heat Transfer, Thermal, Btu per sq ft (hr)	18.37	20.61	27.23	20.44
Hot Surface Coefficient, Btu per sq ft (hr) (deg)	2.11	1.98	2.16	2.13
Cold Surface Coefficient, Btu per sq ft (hr) (deg)	1.44	1.58	1.63	1.58
Transmittance, Btu per sq ft (hr) (deg)	0.300	0.310	0.316	0.320
Conductance, Btu per sq ft (hr) (deg)	0.463	0.478	0.478	0.494
Conductivity, (Btu) (in.) per (sq ft) (hr) (deg)	0.368	0.380	0.380	0.393
Mean Temperature, Surface to Surface, Fahrenheit	41.8	60.5	65.0	78.7
Moisture Content, Per Cent by weight	2.12	1.79	1.79	1.79
Relative Humidity, Hot Air, Per Cent	...	...	...	...
Vapor Pressure, Hot side, mmHg	...	...	...	...
Relative Humidity, Cold Air, Per Cent	...	...	...	...
Vapor Pressure, Cold Side, mmHg	...	...	...	...
Vapor Pressure Difference, mmHg	...	...	...	...
Vapor Transfer, Grains per sq ft (day)	...	...	...	...
Permeability, Grains per sq ft (day) (mmHg)	...	...	...	...

For tests 5 and 6 in Table 1 a saturated solution of magnesium chloride was used. Sodium chloride solution was used for tests 7, 8, and 9; and distilled water was used for tests 10 to 13, inclusive. The relative humidity in the main box was checked by means of a small humidity and temperature recorder. Wet- and dry-bulb thermocouples indicated the relative humidity of the air in the test room.

The sample to be tested was cut to size, 28 x 28 in., and placed in a dryer at 130 F until it attained a constant weight, which was used as the base for determining moisture content.

Readings of thermocouple emf, heater voltages, and watt-hour indications were taken every half hour during normal working hours. This gave consistent temperature and power readings which were averaged for the calculations.

[1] Application of Conventional Vacuum Tubes in Unconventional Circuits, by F. H. Shepard, Jr. (*Proceedings of the Institute of Radio Engineers*, 24: 1574, December, 1936.)

In the dry tests, 1, 2, 3, and 4 of Table 1, there was no vapor transfer from one side of the specimen to the other and the calculations merely involved using the average rate of heat transfer per square foot of test area and the temperature differences across the air films and the specimen to determine the surface coefficients, transmittance, and conductivity. These tests afford an excellent check on the reliability of the apparatus and thermocouples. In each case the calculated conductivity checked within plus or minus one per cent of the results obtained on a similar sample tested in the 12-in. square

HORIZONTAL HEAT FLOWING UPWARD
17 lb/cubic foot)

5	6	7	8	9	10	11	12	13
58.5	81.5	40.6	58.8	79.6	39.4	59.5	79.8	60.0
89.2	109.1	71.0	89.0	109.2	72.3	90.0	110.0	109.6
79.8	100.8	61.2	79.3	100.1	61.9	80.3	100.8	94.6
40.0	64.8	23.3	41.3	62.0	22.2	42.0	61.6	34.1
27.8	53.8	10.1	28.6	49.9	6.5	29.0	49.5	10.3
20.57	20.38	19.40	21.25	23.59	20.62	21.87	24.85	36.62
1.79	3.27	1.80	3.20	4.92	2.18	3.61	5.94	6.13
18.78	17.11	17.60	18.05	18.67	18.44	18.26	18.91	30.49
2.00	2.06	1.80	1.86	2.05	1.77	1.88	2.06	2.03
1.54	1.56	1.33	1.42	1.54	1.17	1.40	1.56	1.28
0.305	0.309	0.289	0.299	0.315	0.280	0.299	0.313	0.307
0.472	0.475	0.464	0.475	0.490	0.464	0.477	0.482	0.504
0.375	0.378	0.369	0.378	0.390	0.369	0.379	0.383	0.401
58.5	82.8	42.3	60.3	81.1	42.1	61.2	81.2	64.4
3.75	1.95	5.86	5.21	4.23	7.33	6.18	4.89	12.0
32.0	34.0	58.0	54.5	47.0	63.0	57.0	52.0	52.0
11.23	21.84	11.27	19.08	30.28	12.79	20.58	34.29	34.28
25.6	17.0	22.0	25.5	13.5	49.0	25.2	15.5	21.6
0.99	1.80	0.39	1.02	1.24	0.77	1.02	1.40	0.40
10.24	20.04	10.88	18.06	29.04	12.02	19.56	32.89	33.88
295.7	544.3	295.7	527.5	821.5	357.8	599.8	992.4	1041.7
28.89	27.16	27.18	29.21	28.29	29.77	30.66	30.17	30.75

guarded hot plates. In one case two 12-in. squares were cut from the sample tested in the guarded hot box. The hot-plate test gave a conductivity of 0.379 Btu per inch per square foot per hour per degree at a mean temperature of 65.0 F; the curve in Fig. 3 gives a value of 0.380 at the same mean temperature.

During the tests in which there was a transfer of vapor the specimen and the main pan of solution were each weighed at intervals of 24 hours or multiples thereof. As far as could be determined opening the box had no effect on the rate of vapor transfer. After the specimen acquired a constant weight all losses in weight from the pan represented moisture transmitted through the sample. In the calculations for these tests it was necessary to correct the total heat transfer for the heat transferred by the water vapor itself. This correction included the heat of vaporization and the heat loss from the vapor as it passed from a warmer to a colder space. The Keenan and Keyes steam tables were used throughout the computations. Where

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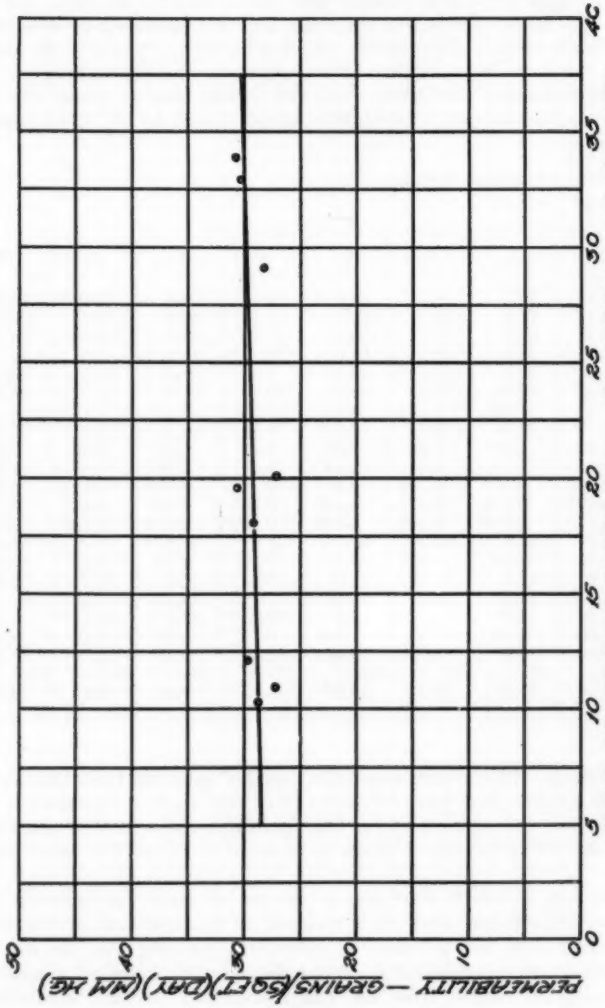

FIG. 4. RELATIONSHIP OF PERMEABILITY TO VAPOR PRESSURE DIFFERENCE FOR 0.795-IN. WOOD FIBER BOARD

evaporation takes place from a saturated salt solution a correction should be made for the heat of dilution. However, calculations using data from Bichowsky and Rossini and the Handbook of Chemistry and Physics indicate that the magnitude of the heat of dilution for the highest vapor rate reported here was 0.04 Btu per (square foot) (hour) in a total transfer of 23.59 Btu per (square foot) (hour) or 0.17 per cent which is less than the accuracy of the other measurements.

The method of calculation employed may be illustrated by presenting the actual calculations for test number 12 in Table 1:

Watt-hour meter reading at 4:00 p.m. 733.40
Watt-hour meter reading at 8:00 a.m. 533.40

200.00

$$\frac{200.00}{8} = 25.00 \text{ (per hour)}$$

25.00×0.994 (constant for meter and conversion of units) = 24.85 Btu per (square foot) (hour)

Vapor transferred was 0.319 pounds in 24 hours, or
0.0133 lb per hour, or
992.4 grains per (square foot) (day)

Internal energy of liquid, u_l, at 110 F = 77.9 Btu per lb
Enthalpy of vapor, h_g, at 49.5 F = 1083.5 Btu per lb
h = 1005.6 Btu per lb

$$\text{Test area} = \frac{18 \times 18}{144} = 2.25 \text{ sq ft}$$

$$\frac{0.0133 \times 1005.6}{2.25} = 5.94 \text{ Btu per (square foot) (hour)}$$

Heat transferred through the specimen due to temperature difference is 24.85 − 5.94 = 18.91 Btu per (square foot) (hour).

$$\text{Warm side surface coefficient} = \frac{18.91}{110.0 - 100.8} = 2.06 \text{ Btu per (square foot) (hour) (degree)}$$

$$\text{Cold side surface coefficient} = \frac{18.91}{61.6 - 49.5} = 1.56 \text{ Btu per (square foot) (hour) (degree)}$$

$$\text{Transmittance} = \frac{18.91}{110.0 - 49.5} = 0.313 \text{ Btu per (square foot) (hour) (degree)}$$

$$\text{Conductance} = \frac{18.91}{100.8 - 61.6} = 0.482 \text{ Btu per (square foot) (hour) (degree)}$$

Conductivity = 0.482×0.795 = 0.383 Btu (inches) per (square foot) (hour) (degree)
Vapor pressure difference = 34.29 − 1.40 = 32.89 mmHg

$$\text{Permeability} = \frac{992.4}{32.89} = 30.17 \text{ Grains per (square foot) (day) (mmHg)}$$

The variation of conductivity with mean temperature is shown in Fig. 3 and the permeability values are plotted against vapor pressure difference in Fig. 4.

All of the tests in Table 1 were made when the specimen was in dynamic balance with the thermal and vapor conditions around it. The vapor flow into the specimen equaled the flow outward. The action of the material when it is dynamically unbalanced with conditions surrounding it may also

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TABLE 2—MOISTURE GAIN OF SAMPLE

DAYS	PER CENT MOISTURE	CONDUCTIVITY	HOT SURFACE COEFFICIENT	COLD SURFACE COEFFICIENT	PERMEABILITY
0	11.1	0.393	1.93	1.18	...
2	13.7	...	...	...	...
5	17.1	0.411	1.89	1.20	...
6	17.6	...	...	...	...
7	18.4	...	...	...	...
8	19.0	0.431	1.93	1.19	...
9	19.8	0.424	1.92	1.16	...
12	21.7	...	...	...	...
13	22.3	0.447	1.92	1.25	...
14	23.2	...	...	...	...
15	24.2	0.460	2.00	1.24	...
16	24.2	...	...	...	...
19	25.1	...	...	...	...
20	25.6	0.466	1.93	1.20	...
21	25.8	0.468	1.94	1.21	...
26	25.8	0.469	1.88	1.25	34.21

be of interest. Moisture can collect in or be driven from the specimen. Tests simulating these conditions were made by subjecting a sample containing 11 per cent moisture to air conditions of 109.0 F and 55 per cent relative humidity on the warm side and 13.5 F and 15 per cent relative humidity on the cold side. The sample gained in moisture content is shown in Table 2 and Fig. 5. The average mean temperature of the specimen during this test period was 66.0 F. Fig. 6 summarizes the data for this specimen at a mean temperature of approximately 66 F. The *balanced* points and tests 3, 11, and 13 clearly indicate that the conductivity is not appreciably affected by the first seven per cent moisture. It is known that the first moisture taken on by a material is adsorbed, a layer of one molecule in thickness covering the surfaces of the fibers. This film does not behave as a free liquid and hence probably has very little effect on the thermal conductivity of the specimen.

TABLE 3—MOISTURE LOSS OF SPECIMEN

DAYS	PER CENT MOISTURE	CONDUCTIVITY	SURFACE COEFFICIENTS	
			Hot	Cold
0	25.8	0.469	1.88	1.25
1	17.6	...	...	...
2	14.7	...	...	...
5	11.1	...	...	...
6	10.5	...	...	...
7	9.9	0.409	...	1.28
8	9.4	...	...	...
9	9.0	...	...	...
12	8.5	...	...	...
13	8.2	0.385	...	1.23
19	7.7	0.378	...	1.34
21	7.5	0.373	...	1.22

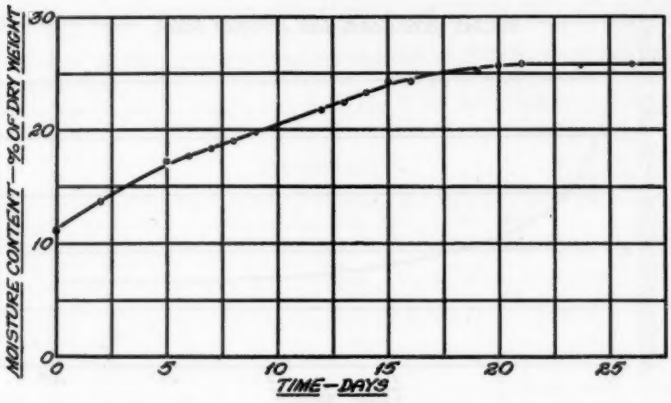

FIG. 5. SHOWING MOISTURE ABSORBED BY 0.795-IN. WOOD FIBER BOARD SUBJECTED TO CONSTANT HUMIDITY CONDITIONS

Test conditions were such that absorbed moisture, or free liquid, did not accumulate in an amount sufficient to affect the conductivity until the 7 per cent point was reached. The *unbalanced* points calculated for the specimen absorbing moisture were expected to be low, since the condensing moisture gives up its latent heat of condensation.

To study the procedure in the reverse direction a piece of ¼ in. plywood, 28 x 28 in., was placed on the warm side of the specimen to act as a vapor barrier. Air temperatures and humidities were held at the same values as before. This caused the specimen to lose moisture, Table 3 and Fig. 7, and

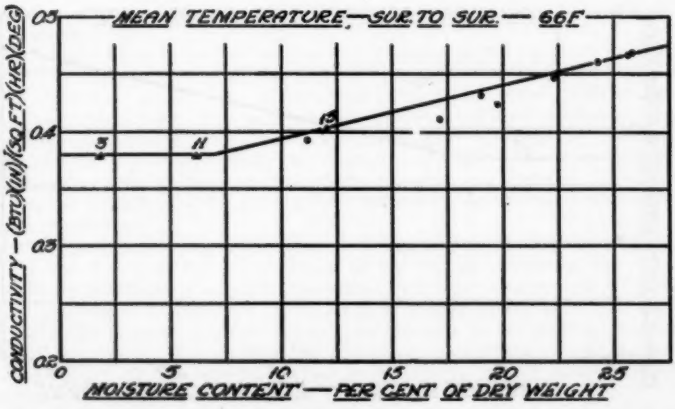

FIG. 6. VARIATION OF THERMAL CONDUCTIVITY WITH MOISTURE CONTENT FOR 0.795-IN. WOOD FIBER BOARD

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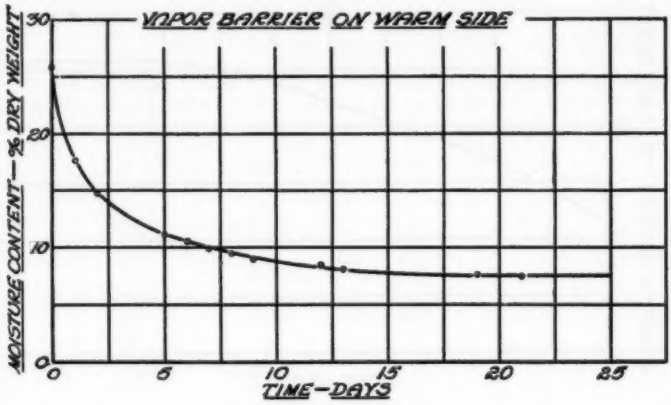

FIG. 7. SHOWING LOSS OF MOISTURE BY 0.795-IN. WOOD FIBER BOARD SUBJECTED TO CONSTANT HUMIDITY CONDITIONS

since the plywood had some thermal resistance the mean temperature of the specimen was reduced to 57 F. During this test the evaporation of the moisture in the sample required heat and consequently the conductivities calculated for the material in the unbalanced state are higher than the actual value. This is shown in Fig. 8 where the curve was drawn by means of the slope of Fig. 6 and the conductivity value at 57 F mean temperature from Fig. 3. The value from test 5 is also plotted here since it helps to define the curve.

The surface coefficients are given in Tables 1, 2, and 3 to show their magnitude and to indicate that they are sensitive to slight changes in surface air

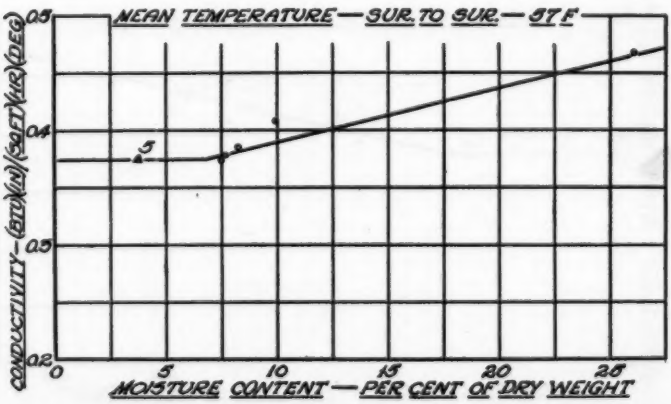

FIG. 8. VARIATION OF THERMAL CONDUCTIVITY WITH MOISTURE CONTENT FOR 0.795-IN. WOOD FIBER BOARD

circulation since the temperature differences across the air films are relatively small.

The use of the test equipment is not limited to testing individual homogeneous materials, but can be used for any material or combination of materials if a representative sample can be made to fit the effective test area of the box.

BIBLIOGRAPHY

1. Observations on the Permeability of Hygroscopic Materials to Water Vapor, by J. D. Babbitt, (*Canadian Journal of Research*, June, 1940).
2. *Thermochemistry of Chemical Substances*, by F. R. Bichowsky, and F. G. Rossini, (Reinhold Publishing Co., New York, 1936).
3. Humidity Measurement, by A. E. Ewell, (*Refrigerating Engineering*, March, 1938).
4. *Handbook of Chemistry and Physics*, 24th Edition, Chemical Rubber Publishing Co., Cleveland, 1940).
5. *Thermodynamic Properties of Steam*, by J. H. Keenan, and F. G. Keyes, John Wiley and Sons, New York, 1936).
6. Moisture Migration, by P. F. McDermott, (*Refrigerating Engineering*, August, 1941).
7. Effect of Moisture on the Heat Transmission in Insulating Materials, by L. F. Miller, (*Refrigerating Engineering*, 14: November, 1927).
8. Theory Covering the Transfer of Vapor Through Materials, by F. B. Rowley, (ASHVE TRANSACTIONS, Vol. 45, 1939).

DISCUSSION

J. D. BABBITT, Ottawa, Canada: I have read the paper with great interest. The authors' idea of measuring the heat transfer and the moisture migration through a material simultaneously is ingenious and their apparatus should lend itself to the investigation of many doubtful points in the theory of moisture migration. The results which they give indicate that the apparatus is capable of considerable accuracy with regard to both thermal and moisture measurements. It is probably the first time that accurate measurements have been made on the influence of moisture transfer on thermal conductance.

I notice with interest that the permeability as given in the last line of Table 1, or in Fig. 4, shows a very small change with mean relative humidity or with vapor pressure difference so that for all practical purposes the vapor transfer can be assumed as proportional to the vapor pressure difference. This lends further support to the assumption that a theory of moisture movement developed on the basis of Fick's Law can be used to evaluate the migration of vapor in any structure if the vapor pressures and the permeabilities of the materials are known. We are thus entitled to use the values of the permeabilities obtained from vapometer measurements for such calculation.

In this respect it is necessary to state that, because of its simplicity and reproducibility, the vapometer test still seems to be the best for evaluating and for comparing the permeabilities of different materials. It is the counterpart of the hotplate apparatus in thermal transmission work. It is in testing the application of permeability coefficients to actual practice that an apparatus such as that used by the authors can be of great value. For instance, an apparatus of this nature could be used to determine if there are any conditions which might arise in practice where Fick's Law would not hold. At present this seems doubtful.

Another useful purpose for which such an apparatus could be used would be to determine the conditions under which condensation would occur within a structure.

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A series of experiments of this nature would give a firm basis on which to set a minimum standard for the permeability of vapor barriers. Finally this apparatus might be adapted to obtain some data on the effect of cracks on the efficiency of vapor barriers—a subject in which we are still at the guessing stage.

E. C. LLOYD, Lancaster, Pa.: The conductivity of insulating material when vapor is present has been for years a moot point with those dealing with the properties of materials used particularly at temperatures below normal. This paper represents an excellent start in the direction of a consideration of the influence of contained moisture passage on the thermal conductivity, and it is to be hoped that additional work will be done which will throw even more light on the subject than is done in this very excellent paper.

Certainly the authors are to be congratulated on the start they have made thus far, and we hope that at a later date we may hear from them with additional information gained in the test equipment developed at the Engineering Experiment Station at the Pennsylvania State College.

L. V. TEESDALE, Madison, Wis.: I am pleased to see this paper and wish I could spare the time to study it carefully at this time, but we are engaged in national defense work and do not have the time to spare on other types of research. The apparatus used seems to be suitable for a variety of tests that should yield new information of real value to those interested in the subject of vapor and heat movement through materials.

Though I have not digested the paper carefully, there are a few points that I would like to discuss briefly. Fig. 4 indicates that vapor movement is proportional to vapor pressure differences. In our studies this principle seems to hold for materials of rather low vapor resistance but not in the case of high vapor resistance. In the latter type of vapor transmission values were not reduced proportionate to reduced vapor pressure differences.

It may be true that on account of the low density of the fiber board its conductivity is not appreciably affected by the first 7 per cent of moisture. In the case of wood having a density of 0.5 at 7 per cent moisture content the calculated conductivity is 10 per cent greater than at 0 per cent moisture content.

JOHN JAMES, New York, N. Y.: In addition to the two major forces of vapor transfer transmitted through building insulation cited by the authors, there is the migration of moisture through porous material due to the influence of convection currents. It is fortunate that the apparatus has been designed so that the box can be rotated through any angular position with the horizontal, as it is probable that varying results of moisture travel will be observed with porous specimens located in a vertical position as contrasted with the horizontal when the heat and moisture flow is from the inside of the box to the outside.

In Memoriam 1942

Name	Joined
George C. Blackmore (*Charter and Life Member*)	1894
Reginald P. Bolton (*Honorary, Presidential, and Life Member*)	1897
John J. Braun	1932
Alice G. Bryant, M.D. (*Life Member*)	1921
Maurice L. Carr	1931
Harrison F. Cox	1929
Francis E. Drescher	1938
Elliott Lee Ellingwood	1930
Phil H. Foster (Sqd. Leader in R.A.F., killed in Action in World War II)	1937
William K. Fox	1939
Andrew B. Galligan	1921
John F. Hale (*Presidential and Life Member*)	1902
Wallace M. Hyman	1920
Herman D. Jackes	1937
Thomas W. Kenney	1937
J. Irvine Lyle (*Presidential and Life Member*)	1911
George E. Mead	1941
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William Ritchie (*Life Member*)	1909
Earl B. Royer	1928
Alfred E. Seelig	1926
Elwood S. White	1921

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